PRODUCT ENGINEERING

VOLUME 7 NUMBER 3

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Coming Issues

Some last minute developments which we wanted to incorporate, impelled us to hold over the article on flexible bellows to the April number. That issue will also contain the concluding installment of Dr. Faragher's article on alloys of aluminum and their uses and the article on the establishment of standards for materials by Hellmund and Miner.

Other forthcoming articles include a series on engineering department practice with reference to the making of drawings by J. K. Olsen, well-known to *P.E.* readers; the application of X-rays to the analysis of grain structures in metals by H. R. Isenburger, the design of V-type leather packings, and detail designs on the application of sealing units.

Skipping now to the June number, this will feature electric and hydraulic drives and controls. Here will be shown how different engineers have applied electric motors and automatic controls for machine drives and how hydraulic units are used for the operation of road machinery, machine tools and other equipment. In each instance, pictures of the machine and diagrams of the application will show exactly how the design was accomplished. Other feature articles in the June number will cover the developments in built-in motors, detailed diagrams for the application of limit switches and functional controls and a reference book sheet for the rapid calculation of the resistance of electrical circuits.

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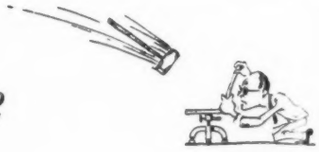
Intimate Correspondence



That January Number

PRODUCT ENGINEERING
NEW YORK

Let me compliment you on that January number, especially the color article by Ketcham. Although selecting colors is not part of my work, I feel that a knowledge of the basis for color selection is of considerable help in designing to get the best looking job. —E.J.S.

Detroit, Mich.

WE thought the same way about it when we commissioned Mr. Ketcham to write the articles.

Don't Get Complicated

PRODUCT ENGINEERING
NEW YORK

Part I of Mr. Van Voorhis' article "How to Make Alignment Charts" was easy for me to understand. I hope that the later articles of this series do not become too complicated for my limited knowledge of higher mathematics. Knowing how to construct such charts saves a lot of time in the engineering department. —D.C.H.

Rochester, N. Y.

THE sole aim of Mr. Van Voorhis in preparing these articles was, to use the author's words "to provide instructions on how to make alignment charts," rather than the development of a theory. We feel certain that D.C.H. will find the successive installment equally easy to understand.

Can't See the Logic

PRODUCT ENGINEERING
NEW YORK

I wish to call your attention to the beautifully composed editorial in February *Product Engineering* under the title "Professions and Unions Can't Mix."

You compare draftsmen and designers with internes and apprentices. I am sure I do not see the logic or else maybe I have been kidding myself for the past fifteen years. If what you say is true then that puts about 95 per cent of the graduate engineers and others in your described class.

There are many draftsmen and designers who are very well satisfied with their profession and due to the magnitude of the work do not feel that they are apprentices, but that they have fulfilled their aspirations.

I will further say that if you make a check of your subscribers you will find the greatest percentage are draftsmen and designers whom you have so grossly insulted. I know that they resent your ridiculous statements to such a degree that they will discontinue your magazine. —P.R.K.

Cincinnati, Ohio

THE statement apparently referred to is "With few exceptions, the job of draftsman is but a stage in a career. It is not a static job but a step in the development to bigger undertakings . . . You might just as well try to unionize hospital internes. Both engineering and medicine are professions and in many respects the draftsman corresponds to the interne."

Maybe P.R.K. has had some unsatisfactory dealings with internes or doctors. Which goes to prove that comparisons are sometimes odious. But in the eyes of the general public, the medical profession is held in high esteem and likewise, we hold the engineering profession in equally high esteem.

A draftsman, as the term is generally understood, is the first step toward becoming a full-fledged designer. His work is limited to tracing and a few minor jobs. Beyond that he is a designer. *Product Engineering* is not edited for draftsmen who consider that they have fulfilled their aspirations and are content to remain static.

Intelligence vs. Education

PRODUCT ENGINEERING
NEW YORK

In your February editorial you ask the question, "Why give up the attempts to organize (unionize) engineers and designers?" and answer it with the statement, "First, they are far above the level of intelligence of the men with which labor organizers generally deal." Only the most conceited of the so-called engineers believes that his intelligence is far above the level of the men in the shop who reduce his ideas on paper to actual machines which perform the work for which they were designed by the engineer.

The mere possession of facts relating to a particular field is not necessarily a sign of great intelligence, nor does the lack of this information prove a lack of intelligence. Unequal opportunities may have given one man an education and denied it to the other, but the intelligence of a man cannot be measured by the quantity of education to which he has been exposed. A white collar cannot be considered a badge of intelligence.

Springfield, Mass.

WE still stand by our statement that as a class engineers are far above the level of the intelligence of the men with which labor organizers generally deal. Insofar as individuals are concerned, we agree that education is not a measure of intelligence. But although there are numerous individual exceptances, we maintain that the level of average intelligence for large groups of people is higher as the education of the group is higher.

At any rate, we will continue editing *Product Engineering* on the assumption that we are writing to an intelligent group of readers who base their judgments and decisions on logical deductions from facts rather than on an illogical appeal to the emotion.

A Professional Agrees

PRODUCT ENGINEERING
NEW YORK

Please accept my congratulations upon your editorial in the February issue of *Product Engineering* entitled "Professions and Unions Can't Mix."

You have covered this subject more thoroughly and with better reasoning than any other article I have ever read along this line. In fact, you have expressed my thoughts exactly. Your comparison of a draftsman with a hospital interne "hits the nail on the head."

I appreciate the spirit in which this article is written, and feel that it is highly constructive. —F.M.Z.

Detroit, Mich.

THIS expresses the views of a man who has come up through the ranks and served successively as designer, chief draftsman, chief engineer and vice-president. One thing is certain. He did not have a union membership to hinder him in his progress.

Names are Easier

PRODUCT ENGINEERING
NEW YORK

On your interesting page "Intimate Correspondence," in your February issue, you title a question to E.J.K., a gentleman from Pittsburgh, "Why Pick on 11 and 13?"

It is surprising that in your answer to E.J.K., you imply that the tin-base babbitt, 89 per cent tin, 7 per cent antimony and 4 per cent copper is non-standard and that the lead-based babbitt 10 per cent tin, 15 per cent antimony, 75 per cent lead is likewise non-standard.

For your information, A.S.T.M. alloy No. 2 formula is: tin 89 per cent, antimony 7.50 per cent, copper 3.50 per cent which, with permissible tolerance, would bring the alloy 89 per cent tin, 7 per cent antimony and 4 per cent copper into the classification as Alloy A.S.T.M. No. 2. As for the lead-based alloy of 10 per cent tin, 15 per cent antimony and 75 per cent lead, this is A.S.T.M. alloy No. 7. Both these alloys are decidedly standard. —H.L., *Metallurgist*

Brooklyn, N. Y.

WE have no disagreement except that alloys should be given as the standard A.S.T.M. or S.A.E. or other authoritative designation, rather than simply chemical composition. Designing engineers, except those who specialize in bearing compositions, are more likely to remember a standard alloy designation rather than the complete chemical composition.

PRODUCT ENGINEERING

GEORGE F. NORDENHOLT, *Editor*

MCGRAW-HILL PUBLISHING CO., INC. MARCH 1936 VOLUME SEVEN, NUMBER THREE

A Technological Boondoggle

ONE of the numerous activities of the Works Progress Administration is a $12,000,000 research program. The first project on this program is a study of the "Re-employment Opportunities and Recent Changes in Industrial Technique." The title of this research project may give an incomplete explanation of its purpose, but its scope is quite well described in the outline accompanying its announcement.

One statement in the announcement reads: ". . . Our analysis starts with the recognition of the fact that the volume of employment in terms of man-hours depends upon the total volume of output and the output per man-hour. . . . Therefore, a national economy with a progressive technology must either contrive to raise the total physical volume of output at a rate which is at least as fast as the increase in the output per man-hour, or prepare to deal with the problem of an ever-increasing number of unemployed." From this statement it may be inferred that machinery is under indictment and will be considered guilty until proved innocent.

An incomplete premise usually leads to an erroneous conclusion. Increased volume of manufactured products invariably creates much employment outside of the factories and plants of the manufacturing industries. Hence, a decreasing ratio of total volume of production to output per man-hour in any one industry is not an index of unemployment. Usually, it results in a greater total employment.

The most familiar example of the effect of greater volume production per man-hour is the automobile industry. As stated by T. J. Watson, president of the International Business Machines Corporation, in his speech before the Machine Tool Congress in Cleveland, "In 1880 the number of livery stable keepers, teamsters, draymen, hustlers, blacksmiths and persons engaged in making carriages, wagons and materials was 717,000. But in 1930 the automotive industry provided jobs for 1,549,000 people." To this number must be added the men engaged in placing automobile insurance, manufacturing and selling gasoline, building roads and bridges, patrolling highways, building public and private garages, and innumerable other activities that must be credited to the low-priced automobile, which was made possible by mass production with labor-saving machinery.

The same line of reasoning can be applied to the manufacture of machinery and equipment for the home. To this development must be credited the employment given to persons engaged in merchandising and servicing the products, in the generation and distribution of power, and in the building of power-houses, generating machinery and distribution lines.

AS FURTHER proof of the soundness of the inference drawn above, we find this question in the announcement: "Has employment in the machinery industry offset the decline in other industries caused by the introduction of new machines?"

The man-hours required to build labor-saving machinery must be less than the number of man-hours saved by the use of such machinery. But take away machinery and civilization will vanish. The question then is: How far toward the vanishing point should we go? Or should we continue the present trend toward an ever higher plane of living for all?

Establishing Standards for Materials—PART I

R. E. HELLMUND, *Chief Engineer* and D. F. MINER, *Manager*
Materials and Process Engineering Department
Westinghouse Electric & Manufacturing Company

Basic principles for determining the most economical number of sizes and grades and showing how the preferred numbers system can be used to establish a definite plan whereby the sizes are chosen on a rational basis

THE evolution of standards for materials and hardware in a large industrial concern is not an easy matter. Any successful plan must take into consideration the varied problems of many plants and frequently of many departments within a single plant. The general individualistic inclination of design engineers, as well as the necessity for their reaching a quick solution of daily commercial problems, tends to lead them away from standards. Therefore, continuous education, vigilance, and supervision by some unit charged with the responsibility for standards become essential and careful planning and supervising of such activities are vitally important to insure the most effectual application of materials at minimum expense.

The Westinghouse company operates fifteen plants manufacturing a wide variety of products ranging from water-wheel generators weighing 1,000 tons to the smallest instrument weighing but a few ounces, and in the merchandising line, from refrigerators to flat irons. After 50 years of experience a plan was developed for promoting materials standards, sufficiently flexible to coordinate the work of the entire company in spite of the wide variety of products manufactured. The following guiding principles have been established on which to base detailed rules of procedure, all aiming to restrict the number of grades and sizes of materials to economic limits and still furnish sufficient choice for satisfactory results.

In general, materials standards must be governed by design requirements and economics. This means that the following requirements must be considered:

1. Satisfactory and sufficient variety (grades) and sizes available to accomplish the engineering result.
2. Avoidance of too many grades and sizes.
3. Avoidance of too few grades and sizes.
4. Rational selection of grades and sizes to provide sufficient range of quality and quantity in economical steps; avoiding grades or sizes too nearly alike.
5. Standards selected in line with national or trade practices to provide minimum cost and maximum availability.
6. Standards coordinated between plants to provide for exchange of stocks or to prevent trouble in moving activity from one plant to another.
7. Adequate information furnished to all departments on standard materials. This aids proper selection from the available materials.

The methods used must be sufficiently flexible to provide for the adoption of new materials and modification of materials to meet new requirements without

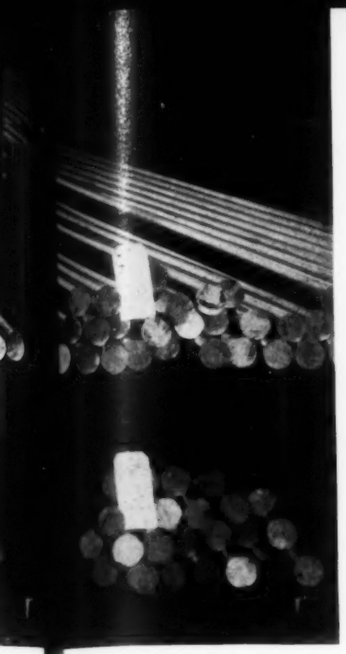
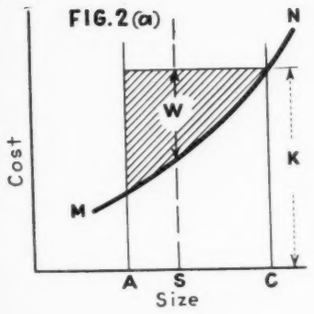
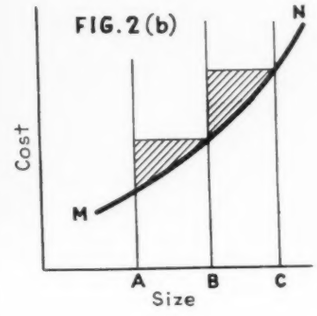
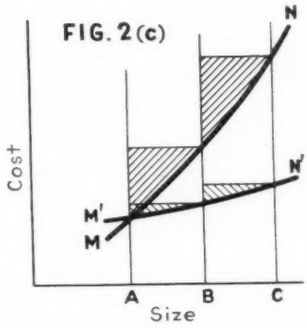
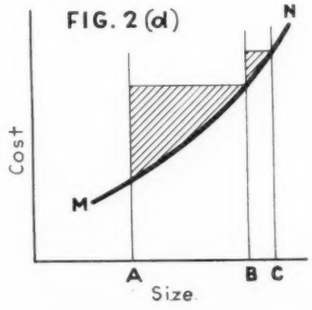


Fig. 1—The sizes of cold finished steel bars follow the fractional series of preferred numbers









Figs. 2(a)(b)(c)(d)—Curves MN represent relation of costs to sizes and shaded areas show cost difference between sizes. At (a), W is cost excess of C over S. Cost difference as in (b) is halved. When curve M'N' is relatively flat, waste is less as at (c). At (d) is result of too closely spaced sizes

fluous grades and sizes of a material is real and appreciable. Furthermore, if the materials are manufactured or processed within the plant, every additional item means expense of tools such as drawing dies for wires; for set-ups in the course of manufacture; or for extra heats and chemical analyses in the foundries. When materials are purchased in small quantities, which is usually unavoidable when there are too many grades and sizes, extra charges are imposed as penalties. Specifically, any amount less than 6,000 lb. of cold-rolled carbon steel strip stock carries a "quantity extra," which for quantities of 150 to 199 pounds amounts to 5.25 cents per pound. It is obvious that the number of materials and sizes should be kept within certain limits.

To illustrate the economical aspect of the above principles, at one plant the storeroom had 99 sizes of carbon steel rod (drill rod) 0.006 in. to 1 in. diameter. Some consecutive sizes differed by only 0.001 in. Since each size was purchased in small quantities, a weeding out of sizes was justified even at the expense of extra machining or change in designs. In another instance, there were 300 varieties of insulated cable, one-third of which were found to be unnecessary.

Conversely, grade and size restrictions should not be carried too far. Too few materials or sizes, with insufficient intermediate steps, will lead to the use of expensive materials when cheaper ones would be satisfactory; or the use of sizes larger than required for intended purpose. This condition is illustrated by Figs. 2(*a*) and 2(*b*). Line *MN* represents the relation of costs to sizes. If there is no standard size between the standard sizes *A* and *C*, it is necessary to use *C* for all intermediate sizes.

When a size *S*, as in Fig. 2(*a*), is required, the standard size *C* with a cost *K* must be used. The line *W* represents the excess cost of *C* over *S*. Assuming that sizes between *A* and *C* are required, the total waste is represented by the shaded area above the curve. However, if an additional standard size *B* is introduced as shown in Fig. 2(*b*), the waste is represented by the two small shaded areas, indicating a total waste of approximately half that shown in Fig. 2(*a*). If cost

Fig. 3—Standardization is responsible for elimination of 200 cable varieties from a former list of 300 kinds

serious duplication of existing grades. Specifically, the system must allow for the following:

1. Addition of promising new grades, first on a trial basis and subsequently in full adoption; also addition of new sizes.

2. Elimination of superseded grades or sizes with minimum cost, confusion and change of records.

3. Simplification or rationalization of lines of materials and sizes existing at any one time.

4. Establishment, as early as possible, of definite plans for groups of materials and sizes to allow for future additions, substitutions, and eliminations.

The expense involved in purchasing, stock-keeping, obsolescence of stock and keeping the records of the super-

Table I—Taper Pin Specifications

Sizes between heavy lines are manufacturer's standards and should be used whenever new sizes are required.
All sizes listed below dotted line require a special reamer having longer flutes than standard.

Number		0	1	2	3	4	5	6	7	8	9	10
Diam. (approx.)		5/32	11/64	3/16	7/32	1/4	19/64	11/32	13/32	1/2	19/32	45/64
Diam. (actual)		0.156	0.172	0.193	0.219	0.250	0.289	0.341	0.409	0.492	0.591	0.706
Lengths—Fractional preferred numbers	3/4	0										
	1	0		0								
	1 1/4 (10 Series)	0	0	0	0	0	0					
	1 1/2	0		0	0	0		0				
	1 3/4			0	0	0	0					
	2 (20 Series)			0	0	0	0	0				
	2 1/4					0	0		0			
	2 1/2				0	0	0	0	0			
	2 3/4					0	0					
	3					0	0	0		0		
	3 1/4 (40 Series)								0			
	3 1/2											
	3 3/4											
	4											
	4 1/4											
	4 1/2											
	4 3/4									0	0	

curve $M'N'$ in Fig. 2(c) is relatively flat, waste will be less than if the cost of the material increases appreciably with size as shown in Fig. 2(c) by the MN curve.

Selection of the number of sizes is resolved into striking the proper balance between the waste (which decreases with an increased number of steps) and expenses (which increase with the number of steps). The most economical selection depends upon the practical importance of the various factors mentioned as well as upon a number of other factors. The magnitude of the development, tool and set-up costs, and costs for the disposal of obsolescent stock of a rapidly changing line is of great importance. Large activity may justify more sizes than small activity.

Many of the factors entering into the problem cannot be fully analyzed because of the lack of adequate cost and statistical data. However, the steps chosen do not have to be those giving the absolute minimum costs. The curve in Fig. 4 represents, for a typical example, the total cost of all factors previously discussed. Near the minimum cost the curve is rather flat, so that a slight departure from the ideal condition usually will not be serious. In general, it will not be difficult, from the data usually available and the use of ordinary judgment, to determine the proper size of steps. For a more complete analysis refer to "Fundamentals of Standardizing Sizes and Ratings as Related to Preferred Numbers," by R. E. Hellmund (*P.E.*, September and October, 1932).

Of importance is the choice of steps that are uniform and avoidance of sizes too close together as illustrated in Fig. 2(d). In this instance, the range between A and C has an additional size B close to C. The shaded areas are but slightly smaller than that of Fig. 2(a) and approximately 25 per cent larger than the shaded areas in Fig. 2(b). With irregularly spaced sizes, all the expenses incident to the three different sizes remain, such as store and record keeping. In addition there is a waste of 25 per cent over Fig. 2(b) by not having the sizes properly spaced. Such conditions will not give the same flexibility to the designer's choice as the uniformly spaced steps.

The advantages of selecting standards in line with

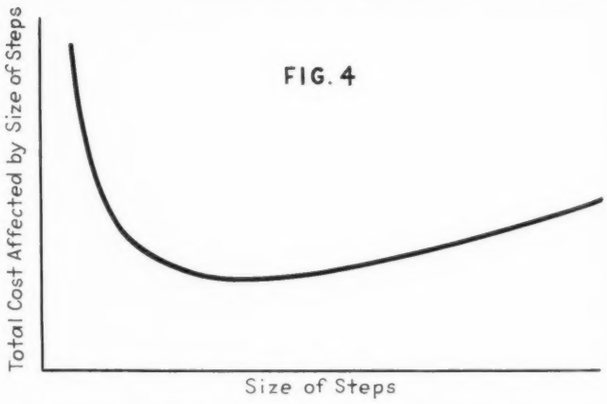


FIG. 4

Fig. 4—Curve shows total cost of development, tools and setup, and obsolescence relative to step sizes

national standards or trade practices are mainly minimum cost and maximum availability. Thus, delivery difficulties and expense are avoided. The Iron and Steel Institute has published standards of quality, tolerances, and other requirements which should be used when possible, since departure from the products listed involves "extras" of various kinds, if tolerances closer than standard are ordered, the price may be increased 25 to 100 per cent. For example, if a carburizing steel requires a grain test, the cost will be 0.9 cents per pound higher than the base price. If a composition check analysis is made, the supplier will charge 0.25 cents per pound extra on the total quantity. Many steels requiring extras for quality, finish, size or tolerance are not stocked regularly by the supplier and therefore cannot be obtained quickly.

Methods for Applying Basic Principles

Some of the previous discussions present convincing evidence of the importance of choosing steps of the proper size and uniformity. With the frequent necessity for adding sizes, desirable results cannot be obtained unless a definite plan is worked out whereby the additional sizes chosen are such that eventually the entire line will be on a rational basis. One of the

greatest aids in such planning is the "preferred numbers" system. Preferred numbers are certain numbers that have been selected by the American Standards Association to be used in preference to any other numbers for standardization purposes of this kind. These numbers give uniform steps, as, for instance in a 1 to 10 range:

The " 5 series" gives 5 numbers approx. 60% apart
The "10 series" gives 10 numbers approx. 25% apart
The "20 series" gives 20 numbers approx. 12% apart
The "40 series" gives 40 numbers approx. 6% apart
The "80 series" gives 80 numbers approx. 3% apart

For any other range, the same numbers are multiplied or divided by 10 or multiples thereof.

The fundamental series are based on decimals, but fractional series also are provided over the limited range where fractions are commonly used for linear dimensions. Supplementary series, having numbers increasing by steps of 9, 18, 40, 100, 150 and 300 per cent can be formed by using certain numbers of the fundamental series. (The system of Preferred Numbers approved as the final standard, is about to be issued by the American Standards Association).

With such a system available, a plan can merely state, for instance, that both dimensions for any new size are to be selected from the 20 series of preferred numbers, using the decimal series up to 0.5 in. and the fractional series for more than 0.5 in. Establishment of a rule for adding sizes accomplishes a two-fold purpose:

1. No sizes will be listed as stock sizes until there is a definite need.
2. Sizes selected over a long period of time will conform to a plan giving uniform steps.

If established trade standards are not based on preferred numbers, other definite rules can be established. In some instances, such a rule might state that from the market materials available, those more closely approximating a certain preferred numbers series should be selected. When preferred numbers or their equivalents are used, it is not necessary to adhere to the same series for the entire range. For example, in determining the proper standard sizes for insulating tubing over a range of 3/16 to 25 in. diameters, it was found that the same ratio of one size to the next could not be retained. The series giving desirable steps between small sizes gave steps too large between large sized tubes. Therefore, the rule adopted for the selection of inside diameters was to follow the 20 series of preferred numbers up to $\frac{3}{8}$ in. diameter, the 40 series from $\frac{3}{8}$ to $\frac{3}{4}$ in. diameter, and the 80 series for sizes greater than $\frac{3}{4}$ in. diameter.

If the possible range of standards is not as extensive as, for example, copper strap, a practice as indicated in Table I can be followed. The sizes kept in stock are indicated by *O*, the blank spaces indicating that for these sizes there has not been sufficient demand to justify their establishment as company standards. Whenever a sufficient demand arises for one or more of these sizes, they can be readily added. A definite plan is most essential. The individual designer or draftsman has many other problems and is likely to select odd new sizes when need arises, but if the selection of the proper size is made convenient he will make a selection in line with the plan.

Table II—Cold-Finished Steel Bars

Material	Use	Cost in %	Machinability in %
Extra free machining Bessemer screw stock	Parts made by screw machines Poor quality	105	120
*Bessemer screw stock	Parts made by screw machines Ordinary quality	100	100
Open-hearth screw stock	Parts made by screw machines Good quality	105	85
*Axle steel (0.35-0.50 carb.)	Shafting	105	77
High carb. steel (ground)	Pins, rods, pivots	700	66
Nitriding steel	Hardened parts	600	71
12 per cent chrome steel	Rust resisting	1,000	55
18-8 steel	Stainless	1,100	45

*Original grades.
Note: Cost and ease of machinability are each based on Bessemer screwstock as 100 per cent.

Rationalization of Material Grades

In standardizing grades of materials it is as desirable to have uniform steps with reference to the essential characteristics as with reference to sizes. The problems involved, however, are not as simple as in the selection of sizes because there are, as a rule, two or more important and conflicting factors involved. Nevertheless with a proper study of the problem it is usually possible to devise a plan which will lead to satisfactory standards.

The list of cold-finished steel bars in Table II shows a more complicated condition. The first four grades have approximately the same cost but different machinability characteristics in reasonable steps. The mechanical properties, not indicated, cover a reasonable range. The last four grades are for specific uses as indicated.

Consolidation of Varieties

In some instances a definite relation of one variety to another of the same type of material can be worked out. There are, for example, many grades of carbon steel castings of different carbon content and different treatments. Physical properties are the governing factors which permit a rational selection to cover the range in desirable steps. A table can be made showing, for example that 0.35 per cent carbon steel castings have a tensile strength of 60,000 lb. per sq. in.; 0.45 per cent carbon of 70,000 lb., and other grades to complete the list.

A combination of surface finish and physical characteristics is found in the problem of rational selection of aluminum sheet grades. Aluminum is available in a number of alloys and in five degrees of temper and two surface finishes. Out of 20 varieties formerly listed, 13 were chosen, which have been found to meet practically all needs. For example, in pure aluminum sheet there are 10 varieties, five tempers in two finishes. Three tempers in one finish were found adequate.

PRODUCT DEVELOPMENTS

• *Modern Steel Press Design*

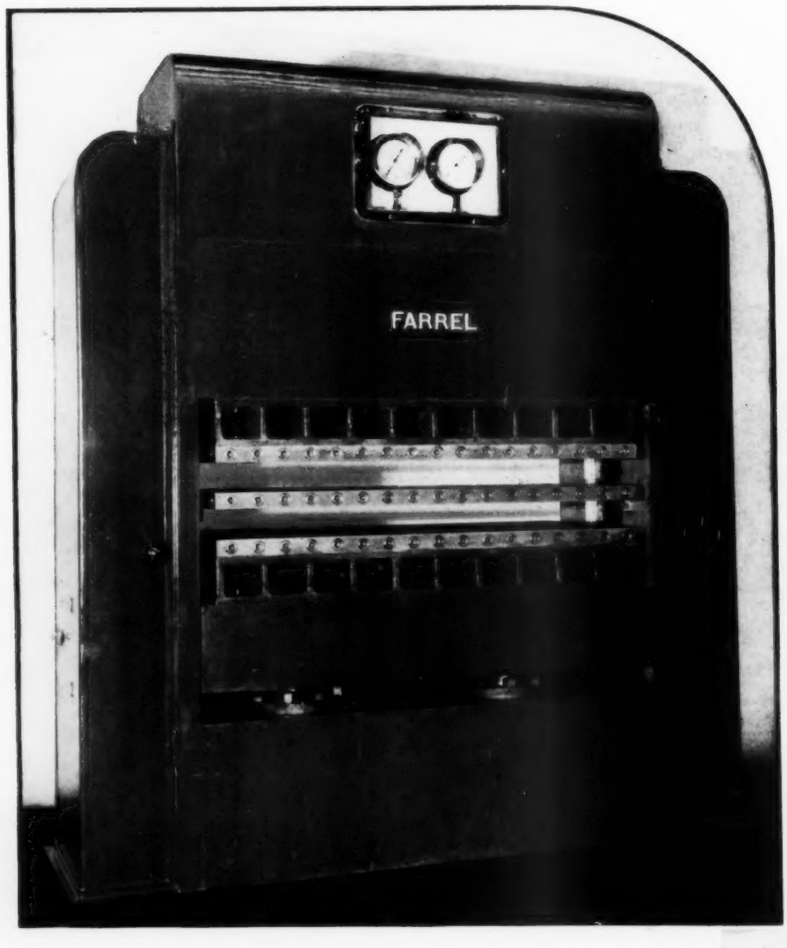
In developing a new line of Farrel-Birmingham presses, the conventional tie rod construction was replaced by side slabs. Two important considerations impelled the change. The fundamental aim was to obtain a structure that was stronger, more rigid, and more readily kept in serviceable condition. Then, in recognition of general tendencies toward better appearance, an entirely new form was sought by the designers.

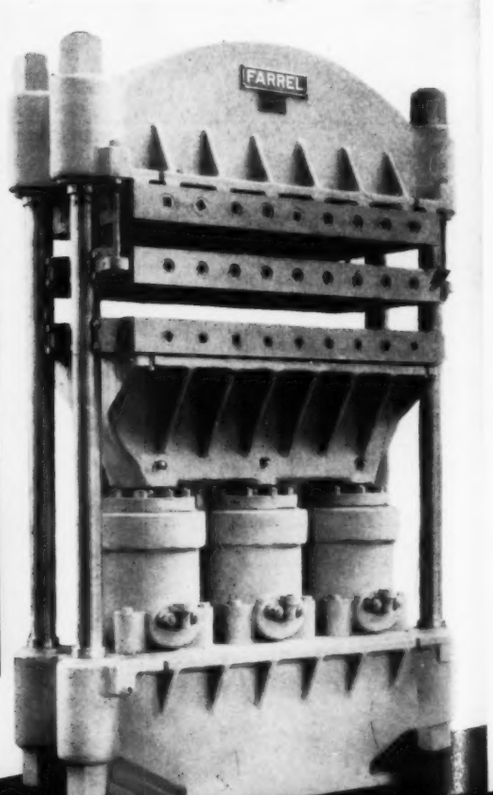
Their success in improving appearance is strikingly evident in the accompanying illustration comparing the new design with the old. Not only are all surfaces smooth and clean, but the form and shape of the new press gives an impression of strength and solidity, which can be attributed to the side-slab design.

The entire tension load is taken by two solid steel slabs, one on each side of the press. Stretch under load, therefore, is more uniform than is possible with four separate tie rods. Top and bottom cross heads are keyed into machined grooves in each slab with a key permitting quick and easy adjustment for any stretch that may occur.

Each press has a self-contained hydraulic power unit. A motor driven variable displacement pump of the radial piston type is mounted over an oil reservoir cored into the top cross head. Only one hand-operated valve is needed to control the operation of the press. Built-in pressure gages are conveniently located in a lighted compartment in the top crosshead. Unsightly piping is both hidden and protected by the welded covers bolted to the side slabs.

Solid side slabs and welded housings in the new line of Farrel-Birmingham presses give an appearance that bespeaks strength and rigidity. Below to the right is shown the old design





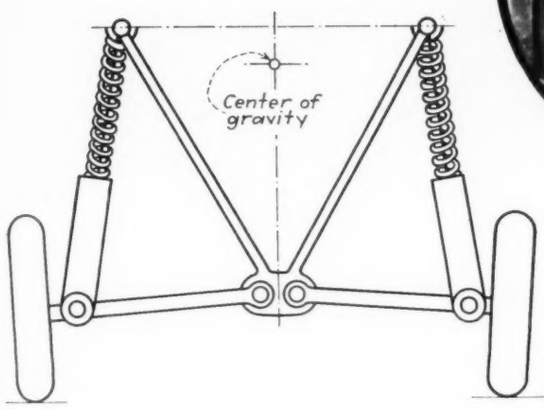
• *Four Point Suspension*

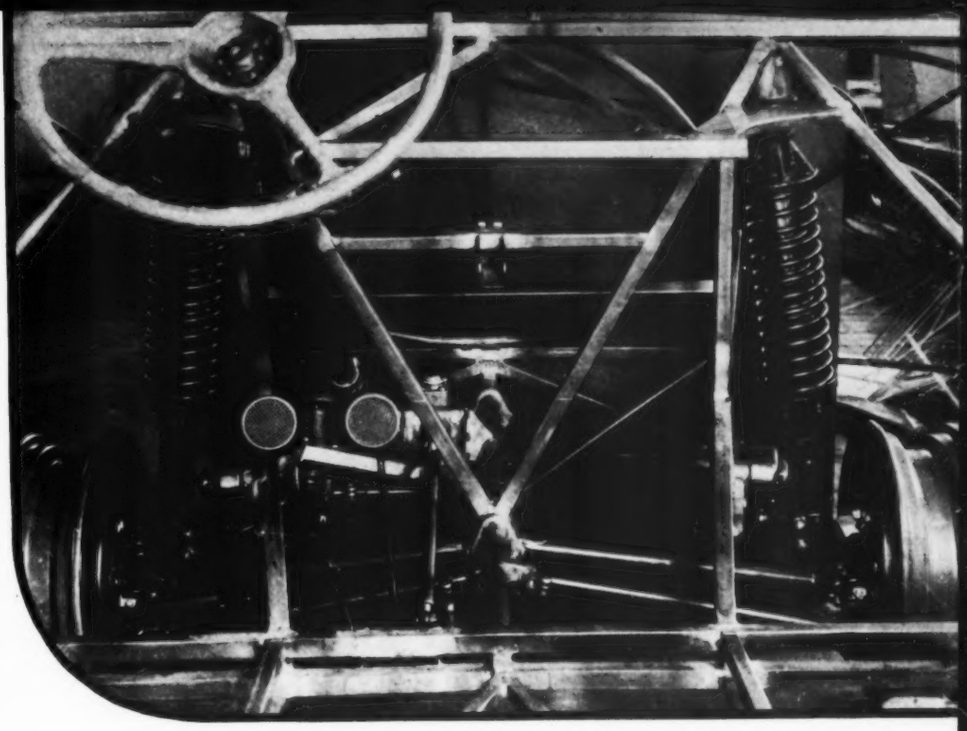
In the design of the new Scarab car developed by William B. Stout of the Stout Motor Car Corporation, all four wheels are independently supported in order to give greater flexibility. Each wheel can rise and fall as required by the contour of the road surface. In order to obtain this flexibility each wheel is suspended from its own support by a spring in conjunction with an "oleo" or airplane type, air-cushioned shock absorber.

As shown in the accompanying diagram, the point of support for each of the four wheels is above the

center of gravity of the car. This results in the car "banking" itself when going around curves. Contrary to the effect found in conventional cars, the body leans inward in going around curves, instead of outward. This is accomplished without the use of torsional side-sway equalizers as used on conventional cars.

As shown by the illustration, the framework of the car is built up of welded rectangular steel tubing.





Each wheel in the Stout car is independently supported at a point above the center of gravity

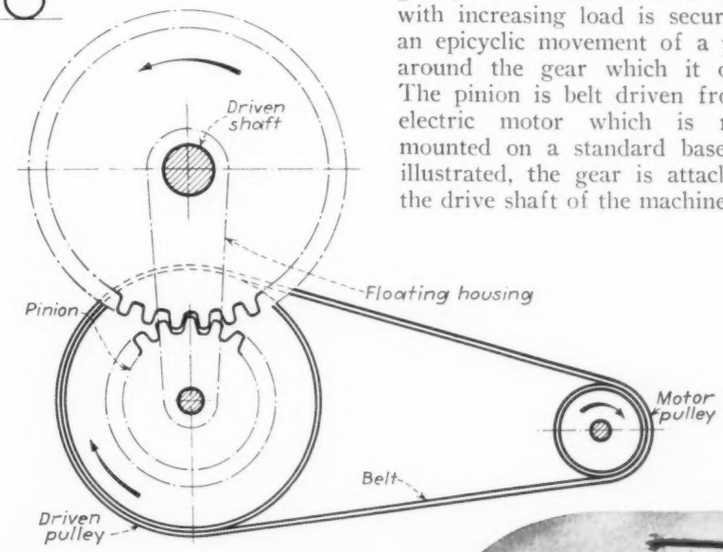
This frame work carries all the structural loads, the body sheets serving no purpose other than inclosing the car.

• *Welded Tubular Booms for Bantam Cranes*

A boom length of 25 ft. was the maximum that could be used with the P&H Bantam-Weight crane when the boom was designed along conventional lines. By replacing riveted plate construction with welded steel tubing, weight was reduced enough to allow the use of a boom 50 ft. in length.

Welding is also used to improve the design of the body of the crane. Frame and side stands are welded together to form a single unit. By the use of a combined box section and X-frame, this unit is said to be three times stronger than an ordinary steel casting of the same weight. Anti-friction bearing mountings for the main shaft and jack shaft are machined from steel billets welded into the frame.

Most of the shafts and gears operate in an oil bath and all other points requiring lubrication are supplied from centralized groups of pressure gun fittings. Power is supplied by a Ford V-8 engine.

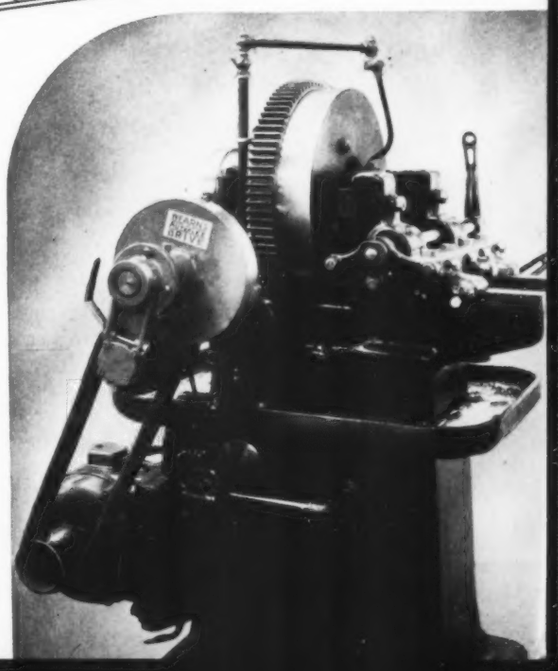


(Above) Pinion reaction is used in the Wearn Autoflex drive to obtain increasing belt tension with increasing load. (Right) The short-center flat belt drive applied to a pipe threader

• *Wearn Drive*

In group drives for production machines, or in other short-center flat belt drives, some method of proportioning belt tension to load is desirable. At the same time, speed reduction is often required, particularly where small high-speed motors are desired in order to save space.

In a compact short-center drive recently developed in New Zealand and now being introduced in England, automatic belt tension is combined with speed reduction. As shown in the accompanying diagram, an increase in belt tension with increasing load is secured by an epicyclic movement of a pinion around the gear which it drives. The pinion is belt driven from an electric motor which is rigidly mounted on a standard base. As illustrated, the gear is attached to the drive shaft of the machine. The



pinion that drives the gear is carried on a frame free to rotate about the axis of the gear. When power is applied to the pinion, it tends to climb the gear and is restrained only by belt tension. As the load increases, the force tending to rotate the pinion about the gear axis increases, thus increasing the belt tension.

In combination with available belt pulley diameters and gear combinations, overall speed reductions as high as 40 to 1 are possible.

In the application to the pipe threading machines shown here the Wearn drive is said to give low belt tension and consequently low bearing loads under idle or light load conditions. Yet when full power is needed from the motor, the belt tension instantly increases to the amount required to deliver the power. The center distance in this drive is only 17 in. Under shock loads or heavy overloads, the belt drive results in reduced impact, yet full power is delivered.

• *Vacuum Switches for Controlling Conveyors*

Accurately and automatically controlled continuous feed is obtained in the new Syntron Gravimetric feeder shown in the accompanying illustration. Material from a hopper is carried by an electromagnetic pulsating feeder to a short belt conveyor driven at a constant speed by a synchronous motor. Control over the feed rate is obtained automatically from the weight of the material on the belt.

The belt conveyor assembly is mounted on one side of the scale shown and is counterbalanced by weights. The rate of feed is controlled by varying the current supplied to the vibrator. Sensitive vacuum switches operated by unbalance in the scale control the current supplied to the vibrator and thus control the rate of feed. These switches require only 0.004 in.-oz. pressure, and instantly speed up or decrease the rate of feed as required to keep a constant weight of material on the moving belt conveyor.

• *Welded Gas Tanks*

Until recently many automobile gasoline tanks were oval in shape with a seam running the length of the tank and around both heads. It was customary to lock seam and solder the longitudinal joint and to solder the heads. However, solder consumption often was excessive, and when tanks were cut open several ounces of solder sometimes were found inside the tank.

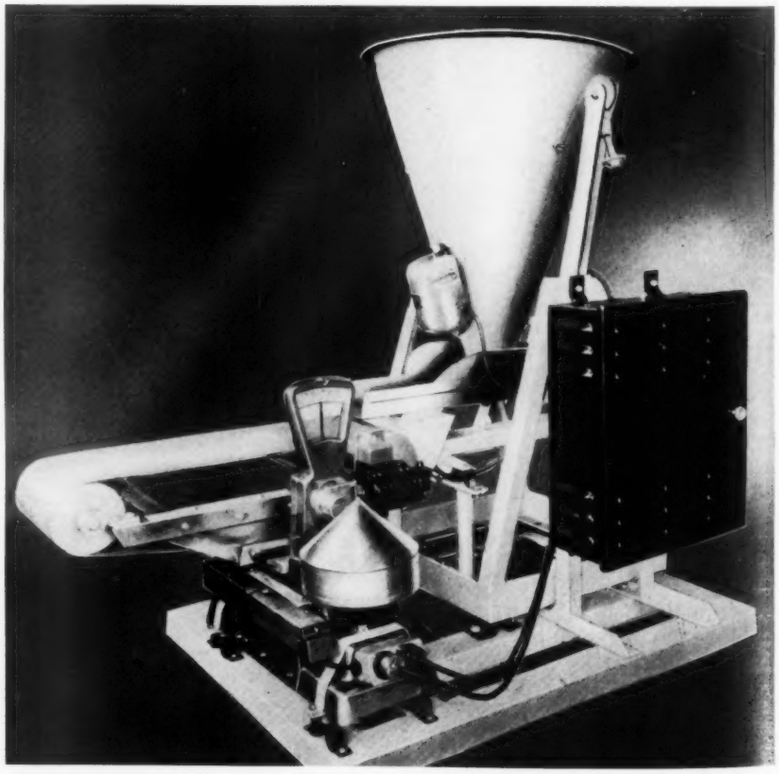
Leaky tanks also presented a problem since it was seldom practical to repair even the smallest leak on the car. Each complaint therefore represented a considerable expense in replacement.

All of these difficulties were cleared up by adopting a new design of tank made up of two halves fitted together like two hats placed brim to brim. The halves were electrically seam welded all the way around. A further improvement developed by Thomson-Gibb engineers was to make the flanges nearly square at the corners so that the weld could be made in a straight line. With this simple change, an automatic conveyor and a pair of seam welders make continuous welds along the sides of a whole string of tanks.

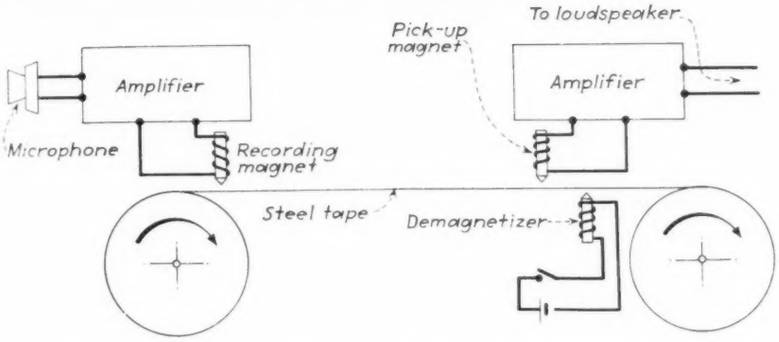
• *The Echo of the Day*

In both England and Germany a new system of spot broadcasting is coming rapidly into use. Whole programs are being given over to news broadcasts from the scene. But rather than placing the broadcast immediately on the air, they are stored until regular broadcasting periods. Several methods of storing the news bulletins have been developed, but the one in widest use today is a system in which speech is recorded on magnetized steel tape. One German broadcasting station has developed a portable equipment which is placed on a truck so that the broadcast can be made directly from the scene. When the truck returns to the broadcasting station the scene is played back from the magnetized steel tape to give a life-like broadcast of the spot news.

The method by which the steel recordings are made is shown in the accompanying diagram. The steel tape is wound from one reel to another, through a magnetizing coil which is energized by the microphone. The steel strip is so magnetized that when it is later passed through a pick-up coil the initial fluctuations in the microphone circuits are duplicated and fed into the



Sensitive vacuum switches operated by scale unbalance hold the rate of feed constant in the Syntron Gravimetric feeder



Spot news for broadcasting over German radio stations is picked up and recorded by magnetizing a steel tape. The stored speech is then taken from the tape and broadcast at frequent intervals

network of broadcasting stations.

The steel tape can be played back as often as desired without losing any of the characteristics of the recording. But when the recording is no longer needed, the tape can be passed through a de-magnetizing coil which leaves the steel tape again ready for another recording.

With tape feeds of about 3 to 4 ft. per sec. only the frequencies between 50 and 2,000 cycles can be reproduced. Thus, the use of this method is limited to speech recording, and music must therefore be recorded in the usual way on wax cylinders or disks. This frequency range is satisfactory for speech.

• *Random Jottings*

For silent operation, a spoke-type fabric gear made by Continental is used on the Hobart meat chopper.

* * *

Fibrous glass thread and yarn manufactured by Owens-Illinois is now being made in white, blue, black and red colors that are fade proof since the colors are neither dyed nor stained surface.

* * *

Cases for industrial recording instruments as large as 14 in. in diameter are now being made of molded Durez. These cases not only stand hard usage, but come from the mold completely finished. In addition, cases can be fluted, ribbed and engraved without appreciable increased cost.

* * *

The Harnischfeger Corporation announces that diesel power has been adopted as standard equipment on their entire line of excavators, ranging from $\frac{5}{8}$ to 4 cu.yd. capacity.

* * *

In a new hammer mill made by Nickel Engineering Works, tool steel cutting edges that are file hard but not brittle are welded to the roll teeth. Their efficient cutting action is said to reduce power consumption to about half that required for ordinary crushers. The roll bearings are interchangeable bronze bushings with separate grease seals.

* * *

Because of its adaptability and its inexhaustible supply, Pyralin, a product of the Du Pont Viscoloid Company is now replacing ivory to a large extent for piano keys.

* * *

To completely exclude dust and grit, the cummutator and brush compartment in the new Black & Decker portable sander is air-sealed. Grease and labyrinth seals protect the gears and bearings.

* * *

Producers of wire-drawing dies are now using K Monel rod casings around the dies, not only to resist corrosion from water and drawing compounds, but also because of the high physical and mechanical properties of this material in its heat-treated condition.

* * *

Because of its corrosive nature, sulphuric acid sludge resulting from gasoline and oil refining operations has always been a difficult material to haul. A truck body fabricated by Gar Wood Industries made of Enduro 18-8-SMO not only resists the corrosive action of the sludge, but the high strength of the steel permits a reduction in the weight of the truck, thereby giving increased carrying capacity.

* * *

Lubricators made by the Detroit Lubricator Company for use on cylinders, cross-heads and pistons in high speed locomotives are now manufactured of bronze containing 1 per cent nickel. This small nickel content is said to cut down rejections because of porosity and lack of pressure tightness.

* * *

The Kewanee Private Utilities Company are now specifying nickel cast iron for all large gears with machine cut teeth.

* * *

Eight spindle speeds are obtained by a complete V-belt drive in the new "Workshop" lathe built by the Southbend Lathe Works. A four-step cone headstock is driven by a single V-belt from the countershaft which also has a four-step V-pulley. An adjustment on the countershaft changes the center distance so that the belt can be shifted and the tension adjusted.

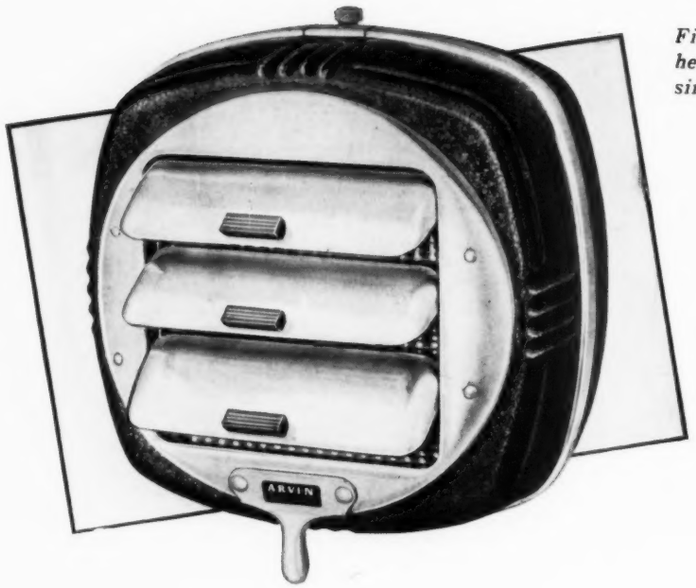


Fig. 1—Each half of Arvin heater case is drawn in a single operation

Flow of Metal in Stampings

EARL C. BOOTH
Chief Engineer, Noblitt-Sparks Industries, Inc.

BECAUSE of the better materials available, possibilities in the use of stampings have increased considerably during the last two or three years. Deep drawing sheets are now being supplied with less surface scale and with better surface finish in general. Stamped parts can be more easily plated and other decorative finishes can now be applied at lower cost. Better rolling practice in the steel mill has resulted in stock of more uniform thickness. With better metallurgical control during the steel manufacturing process sheets now come with much greater uniformity in drawing properties, which decreases scrap losses. At the same time, dies and tools should be designed to take advantage of the better drawing properties in order to decrease the number of draws and increase the life of tools.

In the design of stampings made from these new materials, consideration of the improved drawing properties is necessary all the way through from the initial layout. In making particularly deep draws in one operation, care must be taken to provide sufficient metal to flow into the section being drawn. This may sometimes require the addition of an extra tongue or special corner radius merely for the purpose of providing sufficient metal. Where drastic single draws are desirable, it is sometimes necessary, however, to provide for a second coining or restriking operation to eliminate springback or to get a sharp radius.

In designing the drawn case for the Arvin automobile heater shown in Fig. 1, the deep drawing properties of available steel sheets made it possible for each of the two halves of the shell to be made in a single draw. The design of the shell as well as the dies had to be worked out experimentally. A first approximation of the shape and size of the blank was made by calculating the elongation of the material. With proper allowances for the fact that metal flows readily across a flat sur-

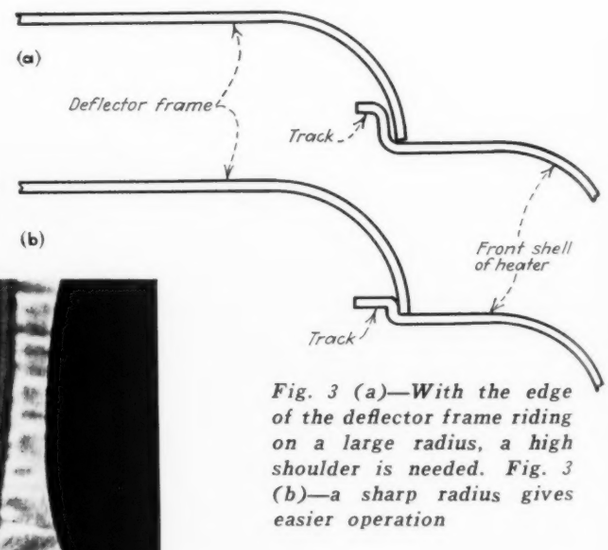


Fig. 3 (a)—With the edge of the deflector frame riding on a large radius, a high shoulder is needed. Fig. 3 (b)—a sharp radius gives easier operation

Fig. 2—In developing designs requiring deep draws, the redistribution of metal can be determined from flow lines and wrinkles

face but flows with difficulty around a corner, the shape and areas of the blank were determined. Then, an experimental stamping was made and the flow lines and wrinkles, as shown in Fig. 2, were studied to determine where the metal was coming from and in what sections more metal was needed.

Metal has a tendency to flow away from concentration points such as corners. The die scratches accurately trace the path of metal in its effort to equalize the pressure of forming. When the dies and pressure pads are extremely smooth, and there are no readily visible scratches, the grain of the metal can be studied to determine the general path of flow.

In the initial design of the single-

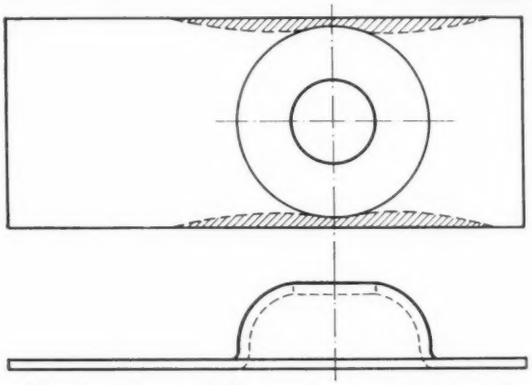


Fig. 5—Increased die pressure on the shaded areas causes metal to draw from the ends. A $\frac{1}{4}$ in. hole in the blank stretches to $\frac{5}{16}$ in.

draw shell a square sheared blank was tried. The piece was somewhat larger than finally used in order to provide increased area for the pressure pad. The operation was not a success, however, because the metal tore at the forming edge of the die. By increasing the radius at the forming edge, tearing and wrinkles were eliminated, but the large radius was undesirable. Finally, the problem was solved by making the blank octagonal and cut close to the corners. The pressure of the holding pad was reduced to allow the metal to flow wherever it pleased. At the forming edge or flowing edge of the die the radius was reduced to a sharp corner as can be seen in Fig. 2. With this new design, the sharp edge did a good job of smoothing out the wrinkles, the light pad pressure and small blank surface prevented tearing, and a smooth shell with negligible scrap resulted.

On another job, a somewhat deeper draw to a similar rounded square outline, a circular blank was used with complete success. This was gradually developed from the octagonal blank. A flange, approximately $\frac{1}{8}$ in. wide was all that was left to be cut off in the trimming die. Last but not least, a complicated blanking die was avoided, since it is easy to make and maintain a die of circular form. It can be turned up on a lathe to accurate clearance, thus doing away with expensive die work.

Fig. 4—Three small parts designed for the same stock thickness are stamped from scrap from the drawn shell

Large radii are usually recommended, but a small radius has the advantage of resulting in less springback. In the design of the deflector frame, a sharp radius was desired to obtain easier operation of the deflector frame. The formed flange at the edge of the deflector frame fits over the outside of a raised well forming a track, and is held in place by a spring tension retainer that bears on the flat inner face of the track.

In a former design, shown in Fig. 3 (*a*), the track was raised and the face of the front shell was extended in order to keep the frame from jumping the track. However, this design had considerable friction and the deflector plate had a tendency to climb over the edge of the track. To avoid tearing the metal during drawing, a large radius was necessary at the break of the track and when the large diameter hole was pierced, the track became wavy.

By making a shallow boss for the track, with both radii as sharp as possible, as shown in Fig. 3 (*b*), the additional strain of the deeper draw was avoided, and the sharp radius did a better job of setting the metal to resist strains already present. Consequently, the track formed was flat and the sharp radius of the outside of the track kept the deflector plate from climbing off or wedging. It also gave a broader flat surface on the inner face of the track which prevents the corners of the retainers from jamming through digging into the inner radius. Because of the easier draw, the new dies outlast dies for the former design.

To reduce scrap, as many parts as possible are designed using one gage of stock. Thus, small parts can be stamped from scrap at the same time that the main stamping is drawn. This not only reduces the number of tools, but also simplifies manufacturing operations in that the exact number of parts required for one set can be made at one time. Thus, as shown in Fig. 4, three stamped parts needed for one heater are punched from the scrap portion of the case stamping.

When it is necessary to have punched holes in parts that are embossed or drawn, it is more convenient usually

to have the holes punched in the blank before the draw. However, distortion in the drawing operation must be considered. For example, in the bracket, shown in Fig. 5, a hole $\frac{3}{4}$ in. in diameter was needed. However, if a hole this size was stamped in the bracket before the boss was drawn, the hole stretched to a much larger diameter. In the example shown, a hole one-quarter inch in diameter originally punched in the stock enlarges to $\frac{5}{8}$ in. diameter in pulling out the boss. Also it was undesirable in this stamping to have the additional material for the boss drawn from the narrow side of the bracket, as shown by the dotted lines. For cost reasons it was desirable to design this part to be made from strip stock. To obtain the straight side necessary for appearance, high work pressure was used at the sides during drawing. Being held at the sides the metal was forced to draw from the ends.

In the design of the die for this bracket the surface of the holder or pressure pad was relieved slightly in the direction of the ends of the stamping where there was plenty of stock to draw from. When the punch starts downward, it tends to stretch the hole, even slightly compressing the metal around its edge. As the draw advances to the point where more metal is needed metal begins to flow from the edge of the hole to the wall of the boss. The hole stretches larger and larger until completion of the draw. In some designs it is possible to develop the original hole with sufficient accuracy so that no further trimming of the hole is required. However, variations in the metal structure and thickness usually makes it desirable, as in this instance, to trim the hole after the draw.

When rivet holes are required in sections that undergo drawing, the holes usually can be punched at lower cost before drawing. But the drawing operation often distorts the holes. Because of the position of the holes in the final pressing of the deflector frame shown in Fig. 6, it would have been difficult to punch the holes after drawing. The outer flange is in the way. The holes, therefore, had to be pierced before forming. With a circular hole to start, the holes assumed an elliptical shape on forming the stamping. Semi-circular holes were therefore put in the blank and they stretched to a circular shape during the drawing operation.

As shown in the illustration below, stampings can also be used to advantage in developing special methods of assembly. The two halves of the heater shell are assembled by four wood screws inserted through the back shell and engaging loops stamped in a frame spot welded to the front shell. Loops shown in Fig. 7 are made for a loose fit on the screw, but when tension load is applied the loops bend in the direction of the pull. Thus, the loops close in on the screw and maintain a tight grip. Even though one of the turns or loops may strip, the remainder will hold. The pressure of the loop against the screw also tends to lock the screw against turning, giving a locking action that increases with the tightening force. A self-locking method of fastening is necessary of course, to keep the assembly tight under conditions of vibration.

Fig. 6—To obtain a round hole in the drawn stamping, a semicircular hole is required in the blank

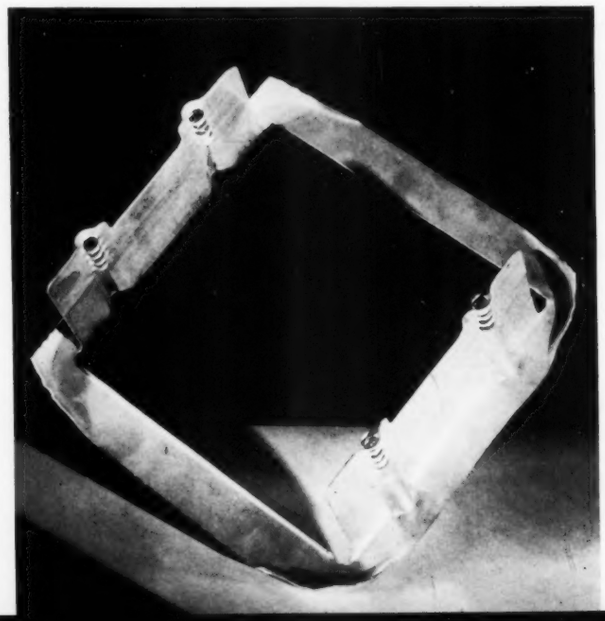


Fig. 7—Long wood screws inserted into the loops secure the two halves of the heater shell. As the screws are tightened, the loops deform and lock the threads

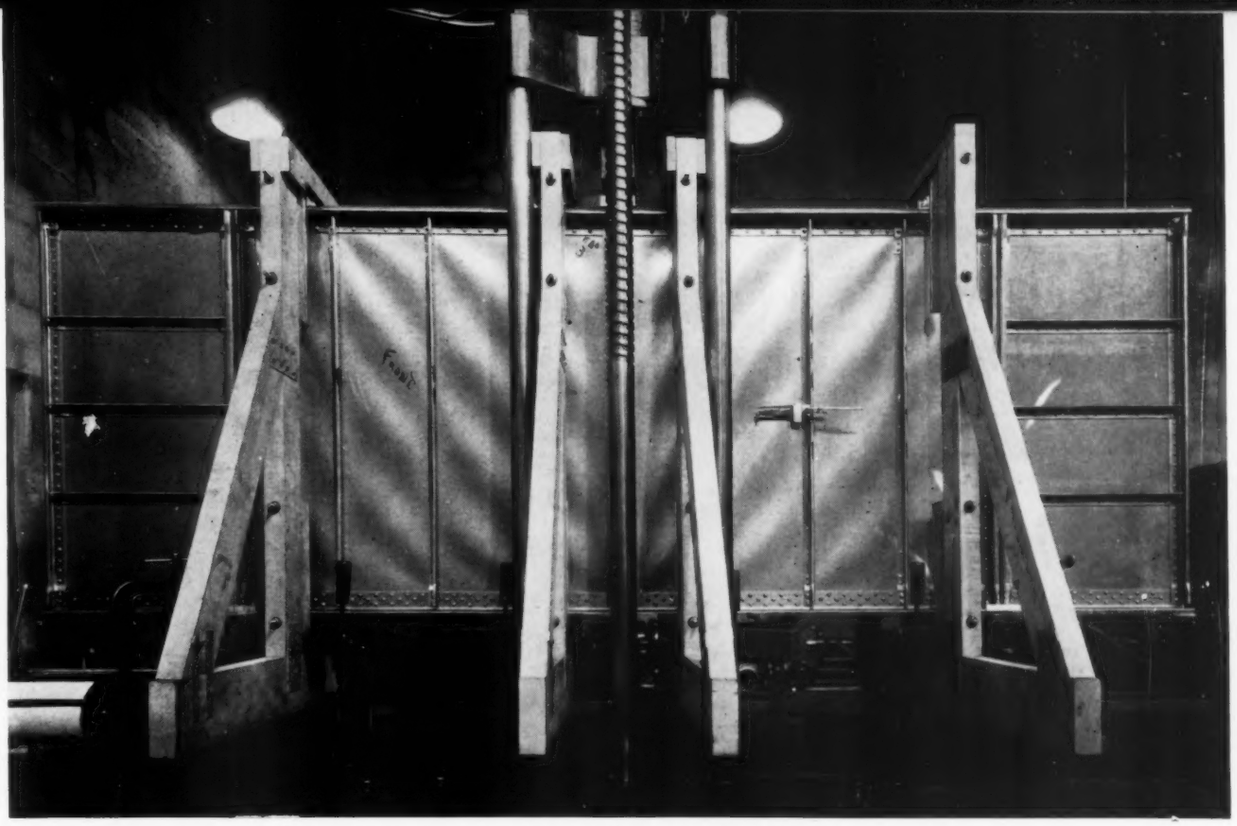


Fig. 1— In testing laboratory of the Glenn L. Martin Company. Note the diagonal wrinkles in the web of this loaded tension field beam

Stress Calculations for Thin Aluminum Alloy Sections

PART II—Design of Shear Members

An analysis of the design of thin plate webs used as shear members, including an explanation of the tension field web and the tapered tension field beams

S. A. KILPATRICK, *Test Engineer*
O. J. SCHAEFER, *Stress Analyst*
The Glenn L. Martin Company

IN LIGHT aluminum alloy construction, the most important shear members used are thin plate webs. The analysis of such members seems to be little understood by designers, even in the aircraft industry where they are encountered constantly. There are two types of webs which need be considered—the shear resisting web and the diagonal tension web.

A thin plate web which is designed to avoid buckling is known as a shear resisting web.

The analytical method of determining the critical shear stress which will cause the web to buckle has been developed by Timoshenko, (see Reference 1 in appended bibliography). This critical stress is given by the equation:

$$f_s = \frac{K\pi^2 D}{b^2 t} \qquad (1)$$

In this equation t is the thickness of the plate, b the length of the short side of the panel, that is, the depth of the panel or the distance between vertical stiffeners, whichever may be the lesser. The coefficient K varies with the aspect ratio of the plate (ratio of length to width) and D is the flexural stiffness factor as determined by the equation:

$$D = \frac{Et^3}{12\left(1 - \frac{1}{m^2}\right)}$$

If we consider 24ST aluminum alloy, for which Young's modulus E is equal to 10,500,000 lb. per sq.in., and Poisson's ratio, $1/m$, equal to 0.25, the expression for the critical shear stress becomes,

$$f_s = \frac{9{,}240{,}000\ K}{(b/t)^2} \qquad (2)$$

The values of K have been determined by Timoshenko (Reference 2) and are plotted in Fig. 2. The coefficient K is here plotted against the ratio b/a, where b, as in Equations (1) and (2) above, is equal to the smaller side of the panel and a is the length

of the larger side. These dimensions are indicated in Fig. 3. If the chord and stiffener angles are on one side of the plate, the dimensions a and b should be measured between rivet center lines. On the other hand, if the web plate is clamped between two chord and stiffener angles, the dimensions a and b can be taken as shown in Fig. 3, between the edges of the structural sections.

If the depth is greater than the width, the terms a and b in Fig. 3 should be transposed. Thus, in Equation (2) b is always the smaller of the two dimensions, and b/a in Fig. 2 will be less than unity.

It should be noted that the formula for critical buckling stress, strictly speaking, is valid only for a panel subjected to pure shear loads. The problem of buckling under combined shear and compression has not yet been solved to the knowledge of the authors of this article. However, the equation seems to give satisfactory results for aircraft structures, probably because the beams in consideration are relatively shallow and the values of K are for laid-on sheet. For the analysis of deep girders, where the maximum compression stress in the web may be quite high because of the combined flexure and shear, the formula must be used judiciously.

Limitations of the Equations

Equations (1) and (2) are of course not valid over the entire range for b/t. If the allowable stress f_s exceeds the shear yield point of the material, the yield point should be taken as the critical stress. For 24ST aluminum alloy, 24,000 lb. per sq.in. is the approximate value for the yield point in shear. It is not advisable to attempt to compute the critical shear stress for thin plates, for example less than 0.032 in. thick, because the actual critical stress is probably lower than a computed value because of the initial unevenness of the sheet. Applying these equations to a typical shear resisting web, such as shown in Fig. 4, the applied unit shearing stress can be calculated as follows:

Assume that $Q/I = 0.1742$, where Q is equal to the statical moment of the portion of the transverse section above the neutral axis taken about the neutral axis. It is computed by taking the summation of the various elements of the section above the neutral axis times their respective centroid distance from the neutral axis.

$$\text{Applied unit shearing stress} = 4{,}000 \times 0.1742/0.064.$$
$$= 10{,}900 \text{ lb. per sq.in.}$$

$$(b/t)^2 = (4.3/0.064)^2 = 4520$$

$$(b/a) = 4.3/36.7 = 0.117$$

From Fig. 2, K corresponding to a value of $b/a = 0.117$ is equal to 5.51. The critical buckling stress can be calculated by Equation (2) as follows:

$$f_s = 5.51 \times 9{,}240{,}000/4520 = 11{,}250 \text{ lb. per sq.in.}$$

Because the applied unit shearing stress is less than the critical buckling stress, this 0.064 in. web will carry a 4,000 lb. shear load without buckling.

It can be seen by inspection of Equation (1) that aluminum alloys are considerably more economical to use than steel for such constructions. Since the modulus of elasticity for steel is approximately three times as large as that for dural, and because the unit weight of dural is only one-third that of steel, a steel panel for a web such as analyzed above will weigh approximately 1.73 times as much as one of dural. This weight comparison between steel and dural is true wherever the allowable stress is proportional to the elastic modulus of material used and the square of the thickness of the sheet.

The vertical stiffeners of a shear resistant web support the web against lateral buckling, and may therefore be compared to beams. The stiffeners carry no axial loads unless a concentrated load is applied at the stiffener. An approximate formula which may be used for computing the required moment of inertia of the stiffener is:

$$I_{ST} = \frac{2.29d}{t}\left(\frac{Vh}{33E}\right)^{4/3}$$

For 24ST. aluminum alloy, this equation becomes,

$$I_{ST} = \frac{2.29d}{t}\left(\frac{Vh}{346{,}500{,}000}\right)^{4/3} \qquad (3)$$

In this equation d is equal to the distance between the stiffeners. The best practice is to make the stiffener thickness equal to that of the web, and then compute the required moment of inertia according to Equation (3) above. For further details, refer to Reference 3 as given at the end of this article.

In aircraft construction, in order to obtain an economical structure, it is usually necessary to allow the sheet to be stressed beyond the critical bucking stress in order to avoid an impractical stiffener spacing. Whenever this occurs, with very thin plates, the sheet wrinkles and assumes a different stress attitude. For the solution of this problem we are indebted to Herbert

***Fig. 2—Critical buckling stress coefficient K** for a thin web plate with edges simply supported as determined by Timoshenko. Fig. 3—Dimensions a and b for chord and stiffener angles on one or both*

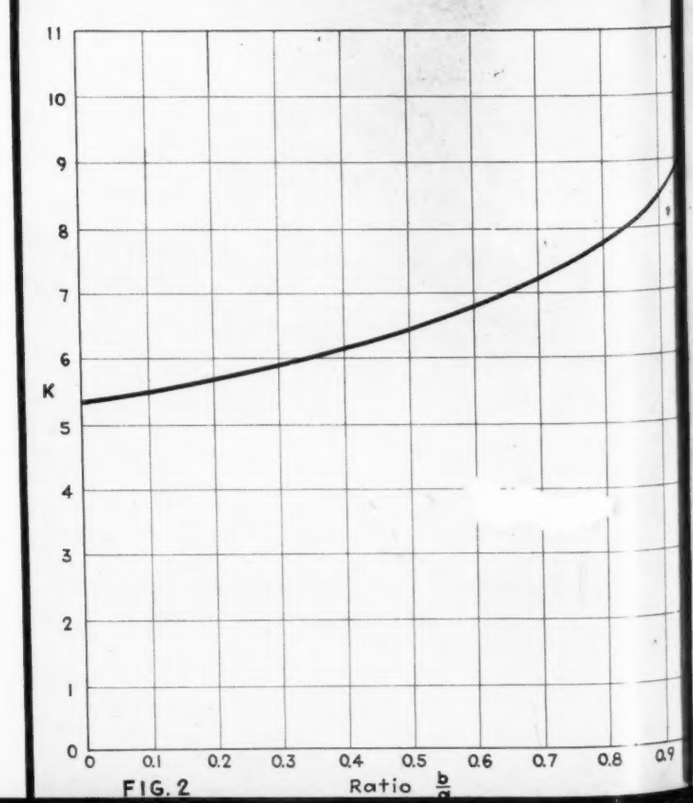


FIG. 2

Wagner. See References 3, 4 and 5 at end of article.

In Fig. 5 is shown a square panel made of rigid sides loaded with a force equal to P. It is assumed that D_1 is a diagonal member capable of carrying only tension loads, but that D_2 can carry some compression. Upon application of a small load such as P, the tension in D_1 will be equal to the compression in D_2. This equal load distribution between members D_1 and D_2 will continue with increasing load P, until the critical buckling load of D_2 is reached. Then, after still further increasing load P, D_2 will carry only its critical buckling load, and the load in D_1 will increase twice as rapidly as before. Eventually the tension diagonal D_1 will carry the major portion of the applied shear P. If we substitute a thin plate for the diagonals, the same line of reasoning applies. In this case, the principal stresses S_T and S_C, equal in magnitude and opposite in sign, act at a 45 deg. angle to the horizontal bars. In a tension field web, the critical bucking stress is so small that the allowable web compression stress, f_s, may be neglected and the entire shear load assumed taken by a diagonal tension field, the tension stress being equal to twice the shear stress.

Before elaborating upon this type of web, a discussion concerning the range of its applicability is in order. The question raised is when should a web be made resistant to shear and when should it be designed to carry the shear loads in diagonal tension? This can be determined by making use of an index value equal to $\sqrt{V}/h$, in which V is the applied shear in pounds and h the depth of the beam in inches. Roughly speaking, if the value of the index is less than 7 the web should be designed as a diagonal tension member; if it is more than 7 a shear resisting web should be used. Since there is no exact dividing line, both types of web members should be examined if the index value is about 7, in order to determine which structure is more economical.

No attempt is made in the following discussion to derive the various equations used since this work has been covered in the literature. See Reference 5. It has been stated previously that the diagonal tension line acts at an angle of 45 deg. with the flanges. This is a theoretical value which is correct for a beam having infinitely rigid parallel chord sections and infinitely rigid vertical stiffeners which are normal to the chord sections. However, the magnitude of the angle for practical design is always close to 45 deg., so this value is sufficiently accurate.

The diagonal tension stress in a tension field web is equal to:

$$S_T = \frac{2V}{ht \sin 2\alpha} \qquad (4)$$

In this equation h should be taken as the distance between the centroids of the chord section. Using a value of 45 deg. for α this equation reduces to:

$$S_T = 2V/ht$$

The allowable tension stress of the web is of course equal to the ultimate strength of the material. Usually this stress cannot be realized because it would require an impractically close rivet spacing for the attachment of the web to the chord sections. Furthermore, at splice points and along the center line of rivets connecting the vertical stiffeners to the web, a reduced web area exists. An allowable stress equal to about 0.7 of the ultimate stress of the material

sides. Fig. 4—Example of a typical shear resisting web. Fig. 5—Beyond the critical buckling load on D_2 the tension in D_1 increases twice as rapidly and eventually it carries the major portion of the load.

Fig. 6—The diagonal lines represent the diagonal field tension in a thin web. Fig. 7—Curve of factors for computing the equivalent free buckling length for tension field stiffeners in terms of total length

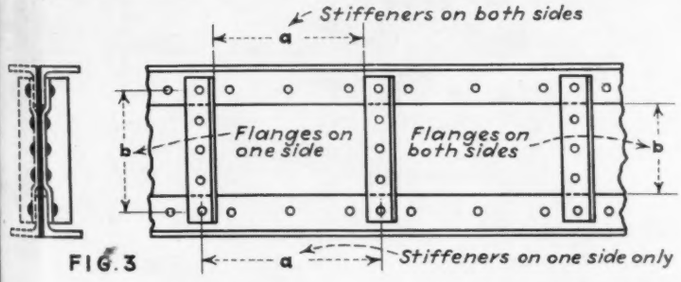
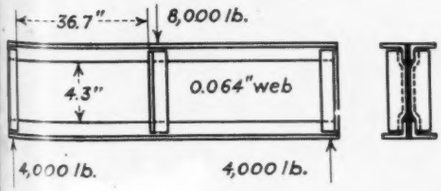


FIG. 3



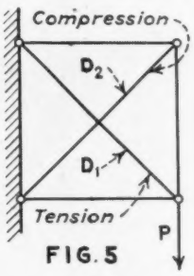


FIG. 5

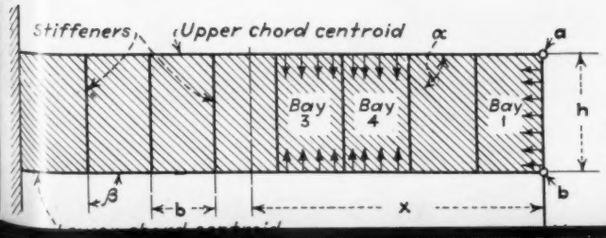


FIG. 6

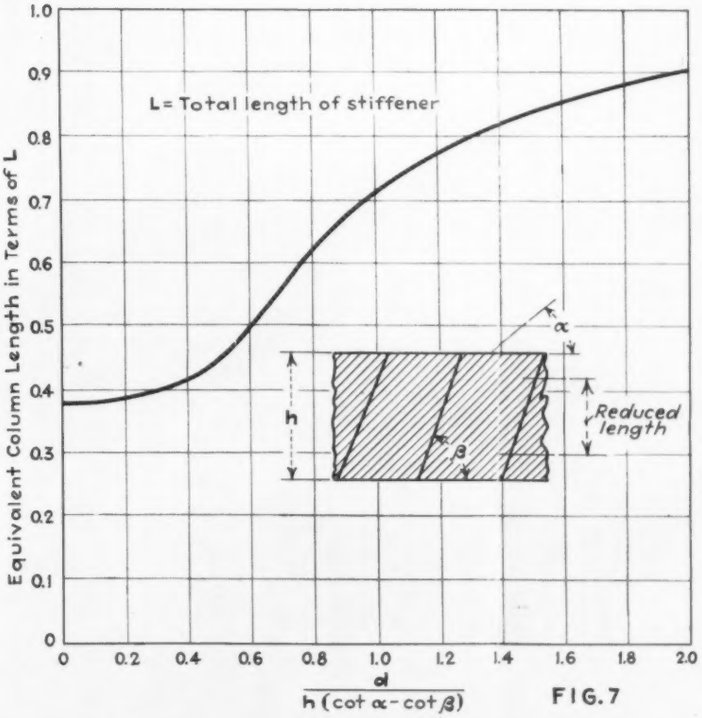


FIG. 7

Equivalent free buckling length for tension field stiffeners is obtained by multiplying the total actual length by the factor from the curve

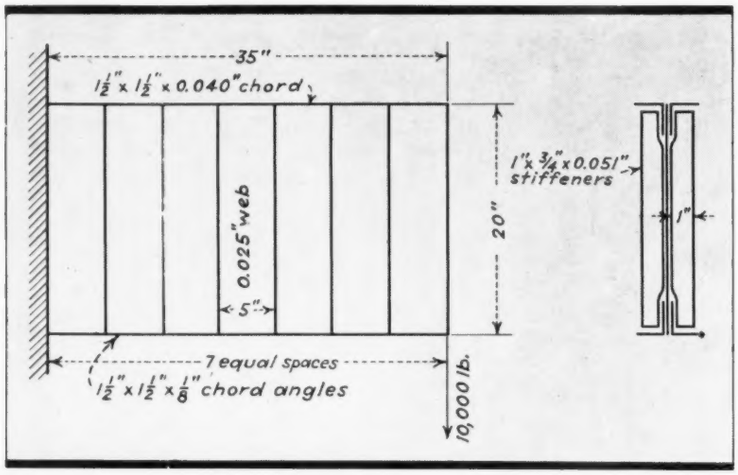


Fig. 8—Example of a tension field web beam, the stresses in which are calculated in the text

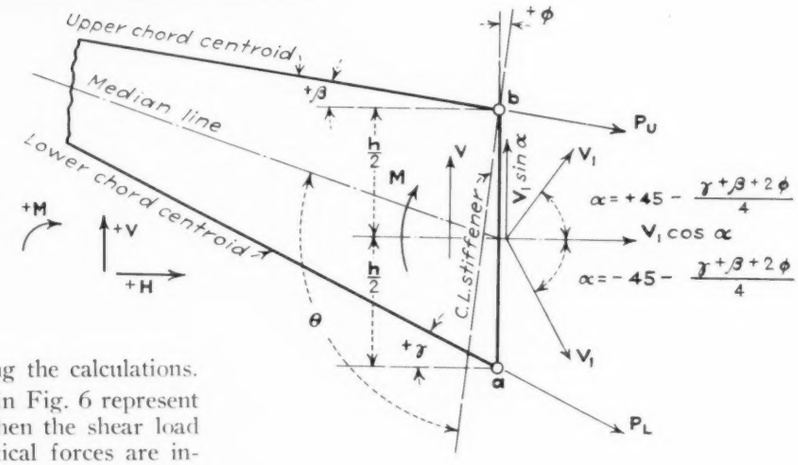


Fig. 9—Portion of a tapered tension field beam cut at section a-b with the equilibrium of forces indicated

is recommended to be used in making the calculations.

The diagonal tension lines shown in Fig. 6 represent the diagonal tension field formed when the shear load V is applied. In bays *3* and *4* vertical forces are indicated, these representing the vertical components of S_T. These componets tend to pull the chord sections together and this action must be restricted by using vertical stiffeners as shown. The chords may then be looked upon as continuous beams in which the points of support are the vertical stiffeners. The compression load in these stiffeners can be determined from:

$$P' = -(Vb/h)\tan\alpha \qquad (5)$$

If α in the above equation is equal to 45 deg., tan will be equal to 1. Equation (5) will then be reduced to:

$$P' = -Vb/h$$

These stiffeners can be made considerably lighter than the vertical members of a truss. It is apparent that if the web is stressed in diagonal tension, any bowing of the vertical stiffener as a result of the column action will cause the web to exert a side force which will tend to push the stiffener back to its original position. For this reason, the stiffeners need not be designed for the full column length, but an equivalent length as shown in Fig. 7. This equivalent length is a function of d/h (cot α — cot β), in which β is the angle between the stiffener and the chord sections as shown in Fig. 6. For vertical stiffeners, assuming α to be equal to 45 deg., the equivalent column length is simply a function of d/h. After determining this column length, the design of a vertical stiffener is the same as for any pin-ended compression member, as explained in Part I of this series of articles, (Feb. *P.E.* pages 49 to 52). Stiffeners must not be spaced farther apart than one-half the beam depth. If this spacing is exceeded, the bending moment in the chords resulting from the vertical component of the diagonal tension field might be excessive. Consequently, the stress lines may concentrate about the stiffener ends.

In bay 1 of Fig. 6, horizontal forces acting on the end stiffener are shown. These forces are caused by the horizontal component of the diagonal tension stress. On a shear resisting web, the chord loads at points *a* and *b*, Fig. 6, are equal to zero since the moment at that section is equal to zero.

In a tension field web beam, the chord loads at points *a* and *b* are equal to $-(V/2) \times \cot\alpha$. For $\alpha = 45$ deg., this term reduces to $-V/2$. Therefore the total chord load at any point at a distance X from the applied load, as indicated in Fig. 6, becomes:

$$\pm M/h - (V/2) \times \cot\alpha \qquad (6)$$

For α equal to 45 deg., this equation reduces to,

$$\pm M/h - V/2$$

In this expression, M is equal to VX. The web is always neglected in computing the section modulus of the beam, since it has no resistance to compression.

Although in the following example the most efficient sections are not utilized, it demonstrates, for purposes of illustration, the procedure for making the computations. Referring to Fig. 8, the material used is assumed to be 24ST aluminum alloy. Using Equation (4) with $\alpha = 45$ deg., the web diagonal tension stress will be:

$$\frac{2 \times 10{,}000}{20 \times 0.025} = 40{,}000 \text{ lb. per sq.in.}$$

An allowable stress of $0.7 \times 62{,}000 = 43{,}000$ lb. per sq.in. will be assumed, 62,000 lb. per sq.in. being the ultimate tensile strength of 24ST aluminum alloy.

Using Equation (6) with $\alpha =$ to 45 deg., the maximum compression in the lower chord will be:

$$\frac{-10{,}000 \times 35}{20} - \frac{10{,}000}{2}$$

$$-17{,}500 - 5000 = -22{,}500 \text{ lb.}$$

Area of the compression chord equals 0.719 sq.in.

$$P/A = \frac{-22{,}500}{0.719} = -31{,}300 \text{ lb. per sq.in}$$

The maximum allowable stress is calculated as follows:

σ = 45,000 lb. per sq.in., the yield point of the material used, in order to calculate the crippling stress
K = 10.8 (assumed)
t = 0.125 in.
$P/A = \sigma \tanh Kt$
= 45,000 tanh (10.8 × 0.125)
= 39,300 lb. per sq.in.

Applying Equation (6) with $\alpha = 45$ deg., the maximum tension in the upper chord is;

$$\frac{10{,}000 \times 35}{20} - \frac{10{,}000}{2}$$

$$= 17{,}500 - 5000 = 12{,}500$$

Tension chord area = 0.237 sq.in.

$$P/A = \frac{12{,}500}{0.237} = 52{,}700 \text{ lb. per sq.in.}$$

Assuming a 15 per cent reduction of area on account of rivet holes, and for an ultimate strength of 62,000 lb. per sq.in. the allowable stress will be:

0.85 × 62,000 = 52,700 lb. per sq.in. allowable stress
By Equation (5) with $\alpha = 45$ deg., we have:
Stiffener load = −(5/20) × 10,000 = −2,500 lb.
Stiffener area = 0.173 sq.in.

$P/A = -2{,}500/0.173 = -14{,}450$ lb. per sq.in.
$d/h = 0.25$

With an actual stiffener length of 20 in., the equivalent length will then be:

$$= 0.39 \times 20 = 7.8 \text{ in.}$$

The factor 0.39 is obtained from the curve, Fig. 7, and corresponds to a value of d/h (cot α — cot β), or simply d/h, of 0.25 for the abscissa.

Radius of gyration of stiffener = 0.443
$L\rho = 7.8/0.443 = 17.6$

Allowable stress = 21,200 lb. per sq.in., obtained as follows:

σ = 45,000
K = 10 (assumed)
t = 0.051
$P/A = 45{,}000 \tanh Kt$
= 45,000 × 0.45
= 21,200 (app.)

The discussion thus far should enable the designer to compute the sizes of members for shear resisting and tension field webs having parallel chord sections and vertical stiffeners. If the beam is tapered, the problem becomes more complex. For tapered shear resisting web, the equations which will give the web shear and chord loads are easily derived, or can be found in textbooks. The applied shear loads in a tapered tension field beam, and the ensuing stiffener and chord loads, are more difficult to determine. Since design formulas for tapered tension field beams have not been published, to the writers' knowledge, it is deemed worth while to present them in this article.

The following formulas are based on the assumption that the chords and stiffeners do not deflect under the action of the diagonal web stress. This assumption implies that the diagonal tension lines bisect the angle θ between the stiffener and median line as shown in Fig. 9, which represents a portion of a tapered tension field beam cut at section a-b so that this section may be put in equilibrium. The symbols used in this figure and the equations are:

V = applied shear at section a-b
M = applied moment at section a-b
P_u = upper chord reaction
P_L = lower chord reaction
$V_2 = V_1 \sin\alpha$ = vertical web shear reaction
$V_3 = V_1 \cos\alpha$ = horizontal web shear reaction
α = angle of diagonal tensile stress with the horizontal
β = angle of upper chord with horizontal
γ = angle of lower chord with horizontal
ϕ = angle of stiffener with the vertical

All other symbols have the same meaning as in the previous discussion.

From Fig. 8 we have:

$$\alpha = 45 - \frac{\gamma + \beta + 2\phi}{4}, \quad \text{(if } V_1 \sin\alpha, \text{ or } V_2 \text{ is positive)} \quad (7)$$

$$\alpha = -45 - \frac{\gamma + \beta + 2\phi}{4}, \quad \text{(if } V_1 \sin\alpha, \text{ or } V_2 \text{ is negative)} \quad (8)$$

$$\Sigma V = V + V_1 \sin\alpha - P_u \sin\beta - P_L \sin\gamma = 0 \quad (9)$$

$$\Sigma H = P_u \cos\beta + P_L \cos\gamma + V_1 \cos\alpha = 0 \quad (10)$$

$$\Sigma M_a = M + hP_u \cos\beta + (h/2) V_1 \cos\alpha = 0 \quad (11)$$

Solving these equations simultaneously we obtain the following result:

$$V_2 = \frac{2}{2 + \cot\alpha\,(\tan\gamma + \tan\beta)}\left[\frac{M}{h}(\tan\gamma - \tan\beta) - V\right] \quad (12)$$

This formula must be used with due regard to signs. If the quantity $[M/h\ (\tan\gamma - \tan\beta) - V]$ is negative Equation (8) must be used in determining α. Conversely, if this quantity is positive, Equation (7) must be used. This method of inspection is necessary since it is impossible for the web to act in diagonal compression.

$$P_u = \frac{-1}{\cos\beta}\left[M/h + \frac{0.5V_2}{\tan\alpha}\right] \quad (13)$$

$$P_L = \frac{1}{\cos\gamma}\left[M/h - \frac{0.5V_2}{\tan\alpha}\right] \quad (14)$$

Having obtained an expression for the web shear reaction V_2 it is relatively simple to show that the diagonal tension stress is equal to:

$$S_T = \frac{2V_2}{ht \sin 2\alpha} \quad (15)$$

And the load in the stiffener is equal to:

$$P' = \frac{2V_2 b \sin^2(\alpha + \phi)}{h \sin 2\alpha} \quad (16)$$

The discussion thus far has dealt with flat shear resisting and tension field webs. For curved shear resisting webs the critical shear stress can be obtained from

$$F_s \text{ (curved sheet)} = \frac{630{,}000T}{R} + \frac{9{,}240{,}000\,K}{(b/t)^2} \quad (17)$$

The first portion of this equation gives the additional stress which a curved sheet can carry above that

of a flat shear resistant web, (Reference 6). The second portion is identical with Equation (2) given in this article.

A solution to the problem of curved tension field beams was recently published. The reader interested in this particular phase of the subject is referred to *N.A.C.A. Technical Memorandum No. 774,* as given in Reference 7 at the end of this article.

There remain two other types of webs which can be classified as shear resisting webs—namely, corrugated sheet, and webs with flanged lightening holes.

It is not worth while to present design formulas to cover these types because the test data used by various investigators is conflicting, and because the present information is so meager that it applies only to special cases, see References 8, 9 and 10.

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(3) Sheet Metal Airplane Construction—Herbert Wagner, Trans. A.S.M.E. 1931, Aeronautical Engineering, Vol. 3-No. 4, Page 153.

(4) Structures of Thin Sheet Metal—H. Wagner, N.A.C.A. Tech. Memo. No. 490.

(5) Flat Sheet Metal Girders with Very Thin Metal Webs—Herbert Wagner. N.A.C.A. Tech. Nemos. Nos. 604, 605, 606.

(6) Structural Design of Metal Airplanes—John E. Younger.

(7) Tension Fields in Originally Curved, Thin Sheets During Shearing Stresses—H. Wagner and W. Ballerstedt. Tech. Memo. No. 774.

(8) The Behaviour Under Shearing Stresses of Duralumin Strip with Round Flanged Holes—Karl Schussler. N.A.C.A. Tech. Memo. No. 759.

(9) An Investigation of Available Information on the Strength Properties of Reinforced Skin Construction—Air Corps Tech. Report No. 3739 (Same as Ref. 2 given for portion of report dealing with tension and compression members).

(10) Shear Strength of Corrugated Aluminum Alloy Sheet—Air Corps Technical Report No. 3682.

Automatic Inspection for Ford Camshafts

A NEW machine which automatically inspects camshafts and not only rejects imperfect shafts but also marks the cams or bearings which cause the rejection, has been designed and built by the Ford Motor Company.

The machine has a gaging accuracy of one ten-thousandths of an inch and makes 25 of the total of 54 inspections to which Ford camshafts are subjected. Except for feeding, the machine is completely automatic in operation and the human element is entirely eliminated in this phase of the inspection.

Distributed among ten stations in the machine are 25 floating gage heads. The camshafts progress from one station to the next, and all gage heads operate simultaneously, so that there are ten camshafts progressing through the machine at all times, each undergoing two or more inspections simultaneously.

As shown in the illustration, the shafts are carried into the machine by an endless linked conveyor, with the length of the shaft cross-wise to the direction of the conveyor. The shafts are placed by hand in registered position in carriers mounted on the conveyor. The left-hand carriers are provided with a key which fits into a locating slot on the front end of each camshaft to register it in the correct position. Two electrical devices at the front end automatically stop the machine if any shaft is not correctly placed.

When the shafts reach the stations, the conveyor stops momentarily. Automatic centers engage the ends of the camshafts and hold them rigid. Then the 25 gage heads lower automatically, measure the shafts, and rise again. As the shafts are moved along to the next station, they are rotated and indexed by a mechanism at the left side of the conveyor to bring them into proper angular position for the next set of gage heads.

The rejecting and marking mechanism is operated by controls within each gage head. The measuring is done by two hard-alloy-tipped pins, one of which is fixed and the other movable. Movement of the pin is mechanically amplified through a system of levers to operate electrical contacts. If the measurement of the shaft varies in either direction beyond the established limits, the magnified movement closes one of two sets of contact points, thereby energizing solenoids. One solenoid actuates a pin on the right carrier which sets the rejection mechanism into motion when that camshaft leaves the machine. The other solenoid contains a marking plunger which is tipped with felt saturated with copper sulphate solution. This daubs the shaft at the point where it is oversize or undersize.

At the outgoing end of the machine are two sets of elevators. One takes the camshafts which have passed inspection of the 25 points to a leather-covered runway where they are passed on for further operations and inspections. The other set of elevators which are put into motion by the pin extending from the carrier containing the rejected shaft lift the imperfect shafts to a high-level rack.

The contacts and solenoids in the machine are operated on a 25-volt circuit to insure long life for the points, and ready inspection of the points is provided by magnifying glasses set into the gage-head housings.

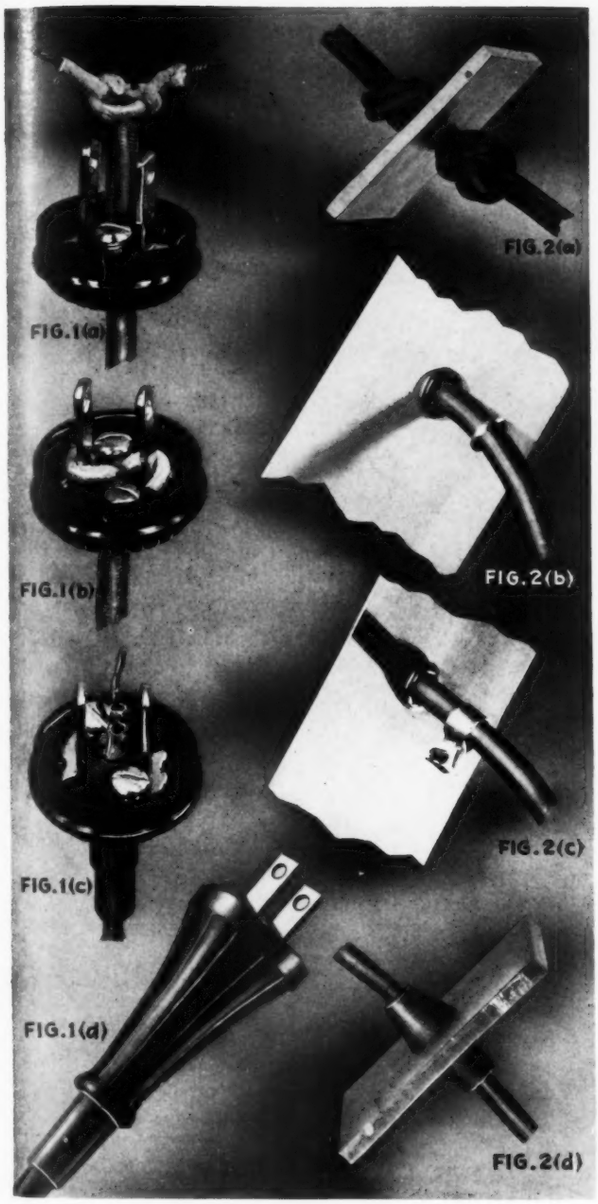


Fig. 1—The "Underwriters knot" at (a) and the loops in (b) both relieve cord strain but require skill to make. The molded rubber strain relief (c) is safer, while the one piece rubber plug at (d) gives the greatest protection. Fig. 2—Knotted cord (a), clamp ring (b) and bolted clamp (c) might possibly be assembled improperly and damage the cord. The molded rubber bushing (d) is simple to apply, anchors the wire and also eliminates chafing

Strain Reliefs and Protectors for Appliance Cords

T. D. FOSTER
General Electric Company

IN the design of electrical appliances, the cord set is a small item. But the appliance is only as good as its electrical connection, for when the cord set fails the appliance is out of service.

Each of the three parts of a cord set—the cord, plug and strain relief, must be selected to meet the service requirements. The cord must carry Underwriters' approval and label. The attachment plug must be of sturdy construction to withstand hard use. But even this is not enough. Of equal importance is the protection afforded the cord at the terminals where the wear is the greatest. A suitable method must be provided to relieve strain at both terminals.

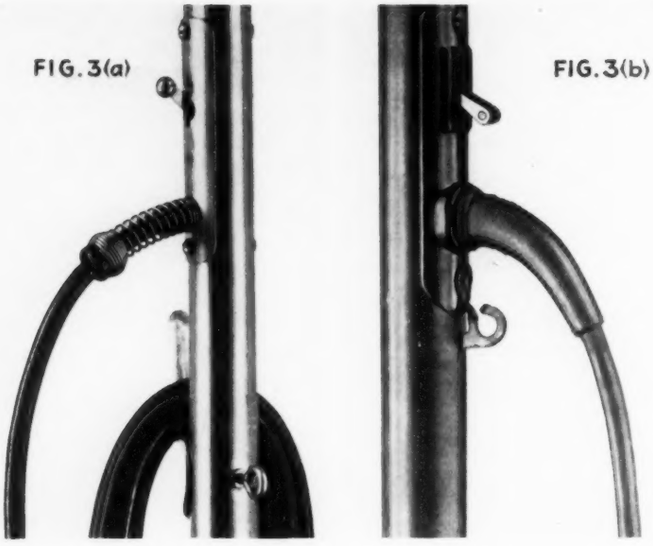
One of the early methods for obtaining strain relief at the plug end of the cord set was the "Underwriters' knot" shown in Fig. 1 (*a*). This knot, fitting into the recess of the attachment cap, relieves the strain on the wires at the binding screw terminals. Another method Fig. 1 (*b*), possible in certain types of plugs, is to take a tight turn around the prongs to the terminal screws and thus relieve strain. However, the efficiency of both of these methods depends upon the skill and care of the assembler.

A third and more satisfactory method that is not dependent on the skill of the assembler is to mold plastic rubber to the cord as shown in Fig. 1(*c*). A knob of rubber shaped to fit the recess in the cap is molded directly to the cord. This is done by vulcanizing plastic rubber under pressure in a mold. This type of strain relief permits the extension of the molded rubber knob in the form of a protecting neck, extending well down the cord. This extension reinforces the cord where flexing and wear is greatest, and gives a permanent strain relief and cord protector.

Molding the entire plug of plastic rubber and vulcanizing to the cord gives a one-piece assembly, as shown in Fig. 1(*d*). This unit thus combines an unbreakable attachment plug, a strain relief and a cord protector.

But the appliance end of the cord as well as the plug end needs protection, for it is just as important that a proper strain relief and cord protector be provided here. In many appliances the greatest wear and strain is exerted where the cord enters the appliance. The cord must be properly anchored within the housing, and where the cord passes through a metal case or plate, a smooth bushing must be provided to prevent chafing.

Of the number of methods for accomplishing the desired results, perhaps the most common one is to tie a knot in the cord itself on each side of the appliance housing wall as shown in Fig. 2(*a*). In some designs a "hog ring" such as in Fig. 2(*b*) is used while other designers provide an elaborate clamp for the cord using a metal band with bolt and nut, as illustrated in Fig. 2(*c*). The effectiveness of all these methods again depends on how well the assembly is



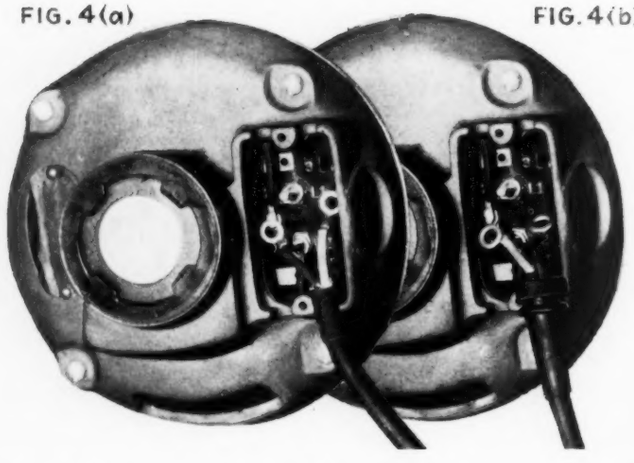


Fig. 3—In vacuum cleaner cords conventional spring protectors (a) require taping to prevent chafing. The molded cord (b) is easier to assemble and requires no taping. Fig. 4—The taped reinforcement on the cord at (a) does not give as much protection as is offered by the soft rubber strain relief (b) vulcanized on the cord

made, the greatest difficulty being that considerable care must be exercised to prevent injury to the cord.

A better method of protecting cords against chafing is to provide a live rubber bushing. In a design recently adopted by a radio receiver manufacturer a small rubber bushing, tapered at both ends, is molded on a flat parallel-conductor cord. The stripped ends of the conductors, extending six or eight inches beyond the rubber bushing, are pulled through a small hole in the chassis, and the tapered bushing is pulled into permanent position, as shown in Fig. 2(*d*). This design prevents chafing and also serves as a strain relief.

To protect the cord at the appliance end, a stiff spring of the type shown in Fig. 3(*a*) is often used. This spring takes up the strain of constant flexing and adds strength to the assembly. However, the cord must be taped at the point where it passes into the appliance.

Modern methods of molding plastic rubber make available a better, simpler and surer way. One manufacturer of vacuum cleaners has now discarded the spring and adopted a rubber sleeve molded to the cord at the point of entrance to the handle, as shown in Fig. 3(*b*). The enlarged rubber section is anchored inside a metal plate by an open washer, and the plate is then screwed into place on the handle. All direct pull on the cord is absorbed at this point, while any twisting or bending of the cord at the handle is absorbed by the strengthened rubber jacket.

By molding the strain relief at an angle the cord can be more easily wound on the cleaner handle. Also, in operation the cord extends more directly from the cleaner handle to the floor, getting the cord out of the way of the operator to a greater extent than is possible with the conventional metal spring.

This same manufacturer now has a rubber bushing molded on the cable connecting the base of the handle to the nozzle and motor housing. This offers a neat and substantial connection to the motor and eliminates any direct pull on motor terminal connections.

In one of the new washing machines the motor cord has a tapered rubber protector molded around and to the cord at the motor end. Thus, the motor terminal housing absorbs the strain before it reaches the points of connection. Without the molded rubber strain relief, the cord must be taped so that the terminal cover clamps the cord, as shown in Fig. 4(*a*). But a more reliable method is to use a molded rubber strain relief on the cord, as shown in Fig. 4(*b*). The enlarged section absorbs the wear from twisting or bending the cord at this point. This one feature has enabled the manufacturer to greatly reduce the number of cord replacements.

Fig. 5—In portable electric tools, soft rubber sleeves molded to the cord gives protection at the point where the cord is subjected to the greatest wear

The same manufacturer has accomplished similar results with molded strain reliefs for heater cords sets on rotary ironers. Molded strain reliefs are also finding wider application to portable electric tools using designs such as the one shown in Fig. 5. Limiting factors such as the cost of changing an expensive die or mold for the housing often require ingenious designing of the rubber cord protectors.

How to Make Alignment Charts

Layout of single and double *Z* charts for equations having three or four functions respectively. Nomograms with parallel or perpendicular solution lines will be shown in the May number

M. G. VAN VOORHIS

PART III

UNDER some circumstances it is desirable to use uniformly divided scales for the division of two functions instead of logarithmic scales. Multiplication of functions can also be accomplished using two uniform scales by rearrangement of the equation. Advantages and disadvantages will become evident in the following procedure for the construction of *Z* charts.

The Z Chart

The *Z* chart provides a means for nomographic solution of multiplication and division of functions in a formula without the necessity for logarithmic scales. Although the type form is shown as a division of functions, it is obvious that the product, xy, may be written $x/(1/y)$ and a reciprocal scale of y used. But in many instances the use of the reciprocal scale is not desirable. The reasons for presenting alternate methods is to avoid reciprocal or other undesirable scales.

Nomogram for $f_1(x)/f_2(y) = f_3(z)$

To lay out a *Z* chart as shown in Fig. 12, draw two parallel lines. Subdivide one line uniformly with a scale for $f_1(x)$ within the limits given and the other line similarly for $f_2(y)$ so that the two scales ascend in opposite directions. The size of each scale is independent of the other and may be as large as the space allows for the given limits. This is shown in Figs. 12 and 13 which are charts for the equation $D = 1.24\sqrt{L/f} + 0.088$.

The function scales are then replaced with scales of the variables in the equation as described under the title "Nomogram for $f_1(x) + f_2(y) = f_3(z)$," page 18, January number. The location of the z scale or reference line is obtained by simply drawing a diagonal line connecting the function values of zero on each scale.

The z scale or reference line in Fig.

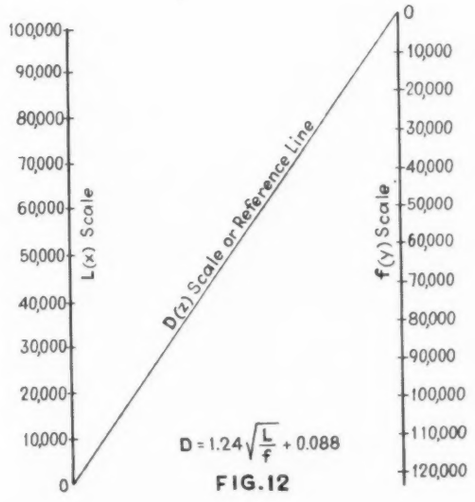


FIG. 12

The Z chart for $f_1(x)$ $f_2(y) = f_3(z)$. Uniformly divided scales ascend in opposite directions. Scale sizes are independent of each other

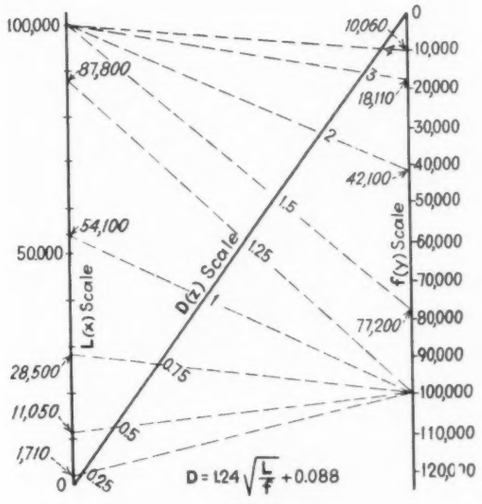


FIG. 13

Method of locating values on the Z or diagonal scale from upper values on both vertical scales

FIG. 14

When zero values are not given, location of diagonal line is computed from similar triangle ratios

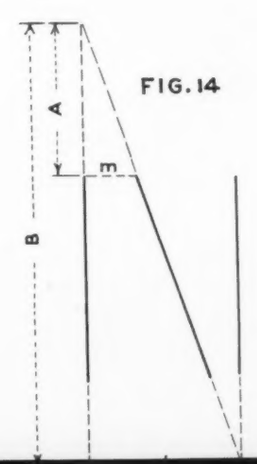


FIG. 14

FIG. 15

In the equation of the type $r/s = (100 - a)/a$ the diagonal scale is uniformly divided as are both vertical scales

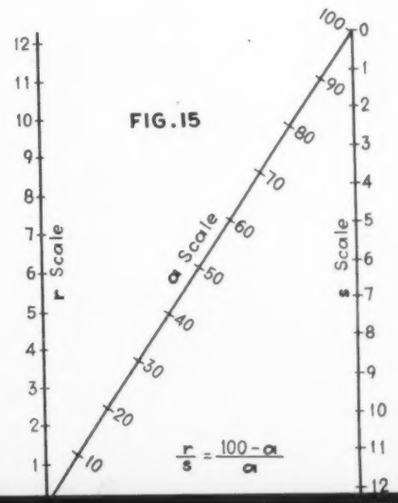


FIG. 15

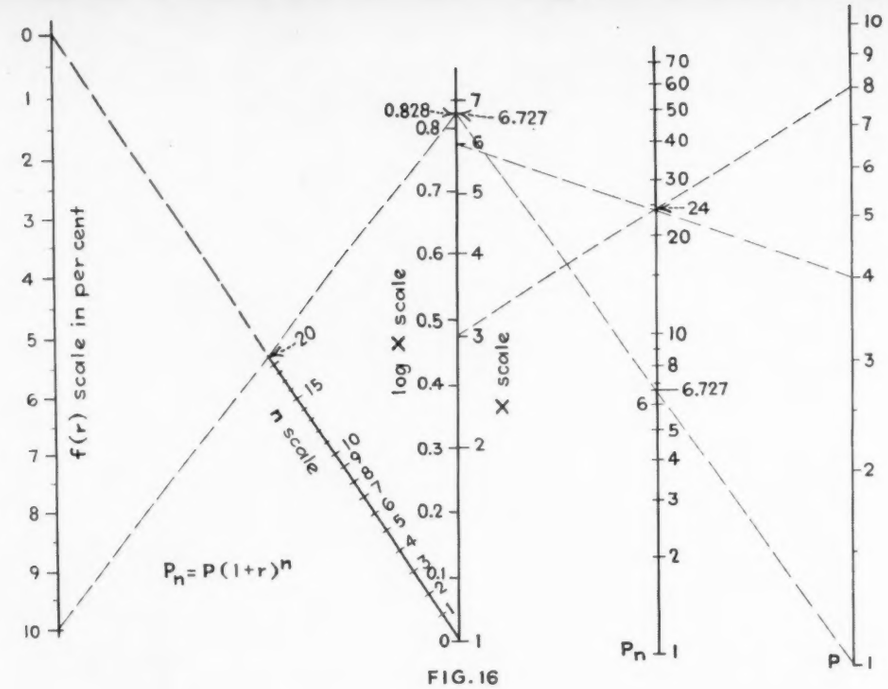


FIG. 16

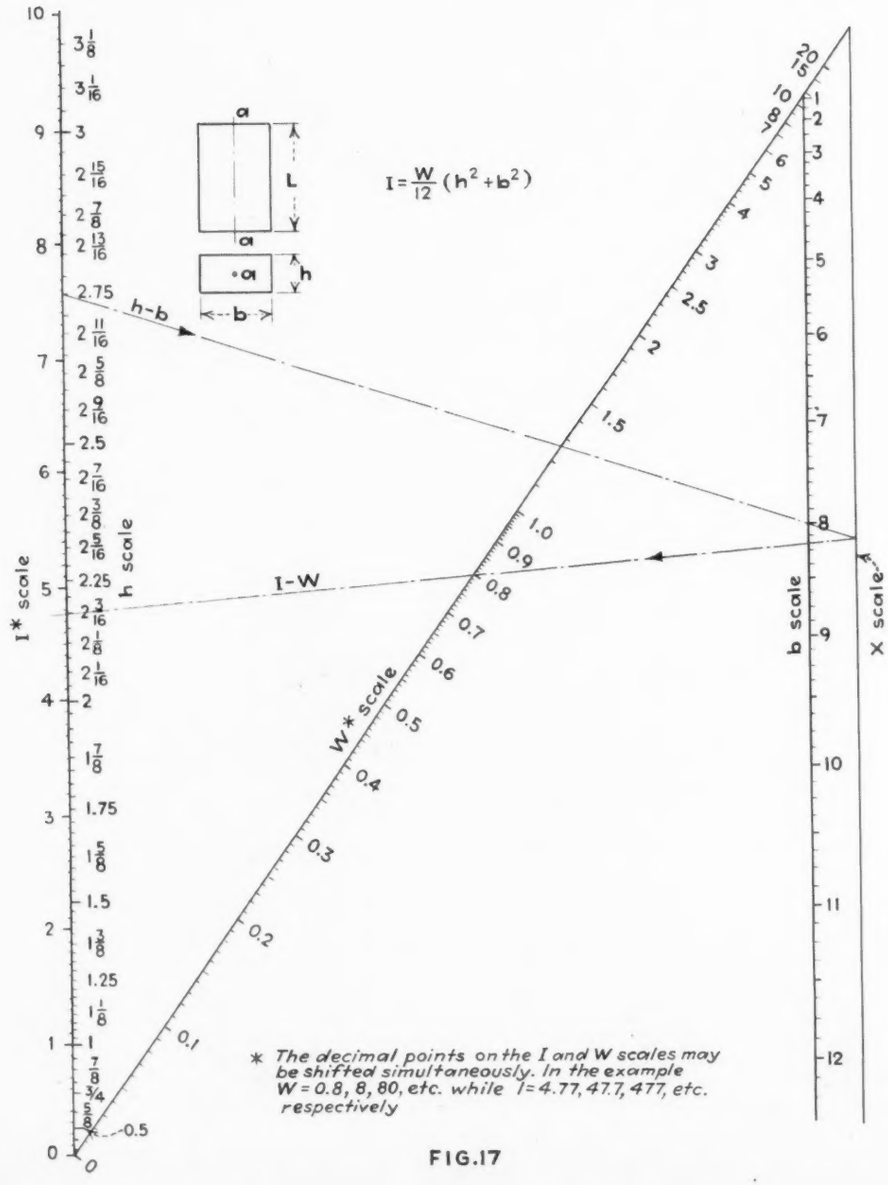

$I = \frac{W}{12}(h^2 + b^2)$
a
L
h
b
h-b
I-W
I* scale
h scale
W* scale
b scale
X scale
* The decimal points on the I and W scales may be shifted simultaneously. In the example W = 0.8, 8, 80, etc. while I = 4.77, 47.7, 477, etc. respectively

FIG. 17

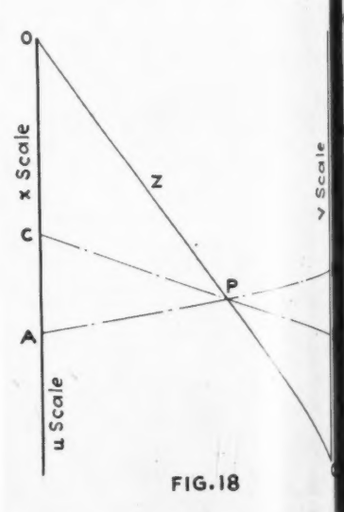


FIG. 18

FIG. 16
Nomogram for obtaining principle P plus compound interest at the end on n years at r rate of interest

FIG. 17
Z chart for determining the moment of inertia of a prism about the a-a axis as shown in the diagram

FIG. 18
Similar triangles can be used to prove the construction of the Z chart as used to prove nomograms having only parallel lines

FIG. 19
Double Z chart for finding relations of forces and dimensions of a toggle joint

FIG. 20
Different type of nomogram for same equation as used in Fig. 19 with two vertical and two horizontal scales

12 may be produced by construction as shown in Fig. 13. Select an upper value of L and solve for f for a series of desired values of D. Locate these values graphically on the diagonal line. Repeat with an upper value of f, solving for L, until as many points as desired are located on the D scale.

If the zero value is not included in the limits of the $f(x)$ and $f(y)$ scales, these scale lines are extended to include the zero value, or the location is computed from similar triangle ratios as in Fig. 14 where $A/B = m/d$. The line extensions are shown in dotted lines.

When the equation $r/s = (100\text{-}a)/a$ occurs, the diagonal scale of the Z chart is uniformly divided as well as both vertical scales. The nomogram for this equation is shown in Fig. 15, representing the resistance r of the slide wire of a Wheatstone bridge with a standard resistance s and a 100-unit length of wire with a slider distance a from one end.

To illustrate the full capacity of the Z chart a nomogram is shown in Fig. 16 for the equation $P_n = P(1+r)^n$ which gives the total P_n for principal (P) plus compound interest at the end of n years at the interest rate r expressed in hundredths.

The equation can be written:

$$\log P_n - \log P = n \log (1 + r)$$

which may be arranged into two equations by introducing the variable function log X as follows:

$$\frac{\log (1 + r)}{\log X} = \frac{1}{n} \text{ and } \log X + \log P = \log P_n$$

The limits are as shown in Fig. 16. The limit for X was chosen as approximately the ratio of P_n/P resulting from 10 per cent interest for 20 years. The function of r $(1+r)$ varies from 1.01 to 1.10. The scale of r is logarithmic between the limits of 1.01 and 1.10 and the scale of X, if drawn, is also logarithmic.

The scales for r and X are drawn on parallel lines so that they ascend in opposite directions; each to as large a scale as the space allows. The diagonal scale line passes through the value of unity on the logarithmic scale of X and zero value on the uniformly divided scale of *log* X and $(1+r)$. The n scale is produced by passing lines through computed values of *log* X and the point 10 on the r scale. The intersections on the diagonal line determine corresponding values of n, the number of years.

The scale of n may be completely laid out by using an equation derived from that relating r, X, and n, in which the selected upper value of $f(r)$ is 1.10. *Log* $X = n$ *log* 1.10. A table of values of *log* X for chosen values of n will assist in locating scale divisions.

n	*log* X	n	*log* X
1	0	11	1.041
2	0.301	12	1.079
3	0.477	13	1.114
4	0.602	14	1.146
5	0.699	15	1.176
6	0.778	16	1.204
7	0.845	17	1.230
8	0.903	18	1.255
9	0.954	19	1.279
10	1.000	20	1.301

The second half of the chart, namely scales P and P_n, is constructed as previously described under the logarithmic type equation shown and explained in Fig. 7, page 53 of the February number. Therefore when P = \$1000; r = 10 per cent interest; and n = 20 years; the principal plus compound interest:

$$P_n = \$1{,}000\,(1 + 0.10)^{20} = \$6{,}727$$

It is obvious that the nomograms in Figs. 1 to 11 inclusive are not adequate to handle the preceding example nor the following equation in solving for the moment of inertia as in the equation:

$$I = \frac{W}{12}(h^2 + b^2)$$

When the equation is divided into two parts, $12\ I/W = x = h^2 + b^2$, the x scale would be logarithmic for one equation and uniformly divided for the other equation. The Z chart as shown in Fig. 17, makes the x scale a uniform scale identical for both equations.

The Double Z Chart

As for parallel line nomograms, similar triangles can be used to prove the construction of the Z chart. In the similar triangles OAP and $O'BP$ shown in Fig. 18,

$$\frac{OA}{OP} = \frac{O'B}{OO' - OP}$$

Letting $x = OA$, $y = O'B$, and $z = OP$, the equation may be written:

$$y/x = (k - z)/z$$

This equation is in the same form as that shown in Fig. 15 and shows why the a scale is uniformly divided.

The double Z chart, however, has a simpler explanation than that for three variables. There are two sets of similar triangles in Fig. 18 having the following relations:

$$\frac{OA}{BO'} = \frac{OP}{PO'} = \frac{OC}{DO'}$$

Letting $x = OA$, $y = BO'$, $u = OC$, and $v = DO'$, the equation

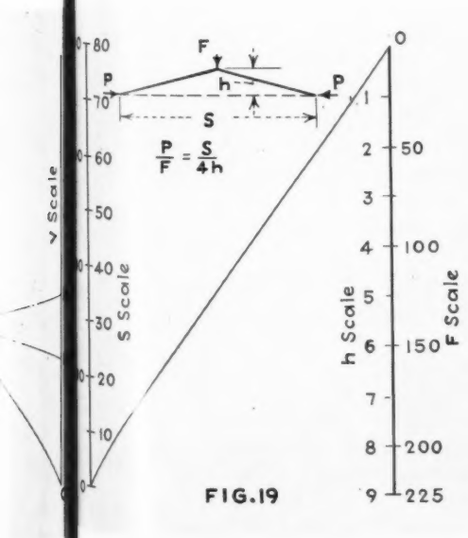


FIG.19

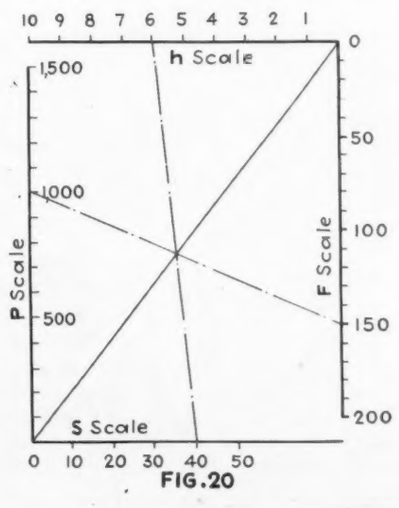


FIG.20

may be written: $x/y = u/v$ as illustrated by the scales in the nomogram.

Nomogram for $f_1(x)/f_2(y) = f_3(u)/f_4(v)$

When laying out a double Z chart, two parallel lines are drawn, one subdivided uniformly with a scale for $f_1(x)$ as in Fig. 19, within the limits given, and the other similarly for $f_2(y)$ so that the two scales ascend in opposite directions. The size of each scale is independent of the other but should be planned in consideration of the following relation:

$$m_x/m_y = m_u/m_v$$

in which m is the length of a function unit. The scales of $f_3(u)$ and $f_4(v)$ should be visualized before determining definitely the size of the $f_1(x)$ and $f_2(y)$ scales.

The u and v scales

Locate on any two parallel lines such that the zeros of the $f_3(u)$ and $f_4(v)$ scales coincide with the zeros of $f_1(x)$ and $f_2(y)$ respectively. The size of the $f_3(u)$ and $f_4(v)$ scale units may be as large as the space allows but the relation between their sizes must be the same as that between the $f_1(x)$ and $f_2(y)$ scales as expressed in the formula given under the diagram in Fig. 19. All function scales are replaced with scales of the variables as described under "Nomogram for $f_1(x) + f_2(y) = f_3(z)$" page 18 of the January number.

Nomograms shown in Figs. 19 and 20 are both for the equation for finding the relations of forces and dimensions of a toggle joint. Scales for P and F in Fig. 19 are drawn first. Using a maximum length for the h scale, the S scale is limited to a shorter space by the given relation between unit lengths. It is usually a matter of trial to obtain the most satisfactory arrangement. This formula works out more satisfactorily with the logarithmic scale type nomogram.

Every different type of nomogram permits new combinations and arrangements which often are valuable merely as a means of making the solution of equations more obvious. The type shown in Fig. 20 is adaptable only to equations of four or more variables. The solution lines instead of intersecting at a common point on a reference line are either parallel or perpendicular to each other and the reference line is eliminated.

The same reasoning applies in Fig. 20 as in Fig. 18 in which the parallel lines make similar triangles with each other. If one more variable were included in Fig. 20 the diagonal line could then be used for one scale. Thus, assuming that h is the product of two variables (xy), an equation could be written: $h/x = y$. The x scale would be opposite the S scale, the y scale on the diagonal, and the line on which the h scale is located would become an unscaled reference line.

Behind the Scenes With the P.A.

G. A. RENARD
Executive Secretary
National Association of Purchasing Agents, Inc.

The editorial in December *Product Engineering* impelled the author to contribute these remarks on what is considered to be a serious problem to the purchasing men, namely, their relationship as the link between the engineering department and the sales engineer.

YOUR editorial "Don't Blame the Purchasing Agent" presents an interesting problem. But I don't believe that any representative purchasing executive would consider placing limitations on contacts between technical experts and his own engineering department. But the purchasing on the other hand, does have the responsibility of separating the desirable and necessary conferences from useless and wasted calls which otherwise would be passed along to the engineering department.

The problem which confronts the purchasing agent is not a simple one and for a long time has been a serious one to purchasing men. As pointed out by Donald G. Clark, purchasing agent of the Brown & Sharpe Manufacturing Company, the problem has been tremendously complicated by this post-depression age. With too few assistants, he finds that more than half of his time is taken up with interviews with salesmen. He also points out that it would be unfeasible or undesirable to save time by cutting down on interviews except during special hours, to adopt the method of offering a discouraging greeting, or to refuse to see all salesmen except those considered desirable to see. Mr. Clark does suggest that 100 per cent frankness with salesmen, seeking their friendly cooperation in a courteous way, discouraging practices that lead to too frequent interviews, and discouraging in a courteous way the purely social aspects of the call, will do much toward saving time.

A properly qualified sales engineer having a specific service or material to present to a company that should be a prospective user, should not make haphazard calls. His position and his proposition should merit greater preparation in planning than a salesman calling regularly on the trade. If his knowledge and his proposal warrant it, surely some advanced arrangement would be made if any value is to be placed on either his own or his prospective customer's time.

There is a growing tendency in many business organizations where technical information and knowledge are essential to progress and development, to use technically trained men in the purchasing organization, either directly or as purchasing engineers who brings the purchasing and engineering department into close cooperation and coordination.

In companies where technically trained men are in the purchasing organization, fewer sales conferences with the engineering department are necessary. These men are in an excellent position to judge whether or not there would be anything gained by having the sales engineer interview the engineering department. It is surprising how many salesmen interviews are unimportant, and the purchasing department is rendering a genuine service in protecting the chief engineer from salesmen who have nothing new to offer.

Lubrication of Sleeve Bearings

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Bunting Brass & Bronze Company

Methods of grooving bearings to distribute lubricant and to prevent end leakage, designed to avoid reducing the load-carrying capacity of the bearing

PART IV

PREVIOUS articles demonstrated that internal grooves are not only ineffective for distributing the lubricant in a bearing, but that the grooves also cut down the load-carrying capacity by interrupting the lubricating film. For distributing lubricant, bearing grooves should be placed on the outside of the bearing with the oil delivered to the desired point through holes in the bearing wall. Internal bearing grooves should be used only to retard end leakage.

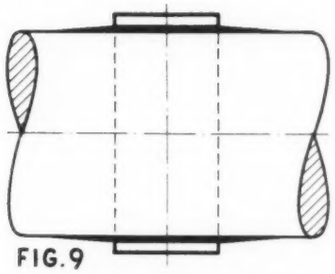


FIG. 9
Sharp edges allow the oil to leak along shaft

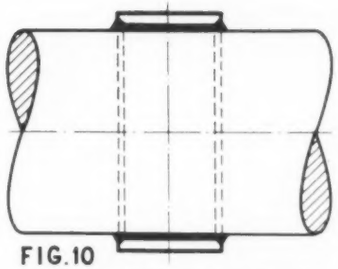


FIG. 10
Chamfered edges form beads of oil, reducing leakage

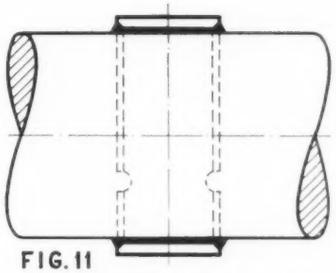


FIG. 11
Notches collect the oil and return it to a sump

Fig. 9—Capillary action and film pressure cause end leakage in plain bearings. Fig. 10—Chamfering the bore causes formation of a bead of oil that retards leakage. Fig. 11—Notches in the chamfered bearing collect oil from the bead and return the lubricant to a sump

As illustrated in Fig. 9, the combined effect of capillary attraction and the film pressure will cause the oil to run along the shaft. Leakage is greater as the high-pressure area is approached and in bearings with sharp edges there will be nothing to prevent this flow. A simple method of retarding leakage, although it might not prevent it entirely, is to chamfer the edges of the bearing as shown in Fig. 10. Capillary attraction will cause a bead of oil to form, thereby retarding the creep of oil along the shaft. Chamfering also has the advantage that the shaft can be easier assembled into the bearing. The reduction in bearing area resulting from chamfering usually is insignificant.

Fig. 11 shows how notches in the bearing end which has been chamfered will serve to collect the bead of oil that is formed. These notches register with return holes in the housing of the bearing, and conduct the oil to a sump or well.

If drains or returns are desired, a notch at each end of the bearing outside of the high-pressure area will provide a path for the return of the oil to the sump. In most instances, it is better from a manufacturing standpoint to put this notch at the bottom of the bearing. This method is shown in Fig. 12 for a wool-packed bearing. Such a construction can be used next to a flange or shoulder on the shaft, or on any type of sleeve bearing. When next to a shoulder, the notch should be made deeper, to prevent the shoulder from acting as a slinger. In any design, the oil drain should be placed so that it will remove not only the oil from the notch, but also any oil that might be thrown off the shoulder. Separate oil slingers are also effective, but they add extra expense and in some instances may be a source of operating noise.

Another method of placing oil return holes is shown in Fig. 13. This is a wick lubricated design. This bearing could be improved by omitting the circular

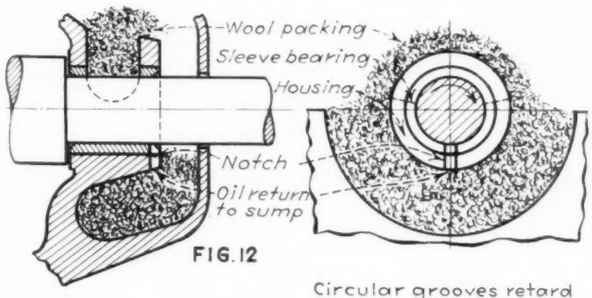


FIG. 12

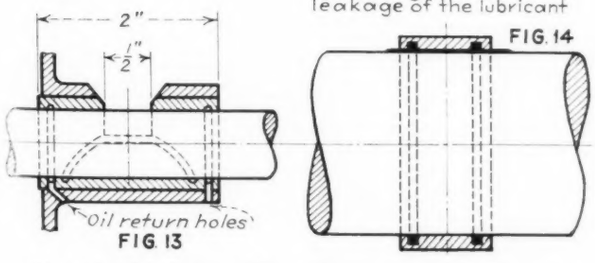


FIG. 13

Circular grooves retard leakage of the lubricant

FIG. 14

Fig. 12—Radial notches return excess oil to the sump. Fig. 13—The length of this bearing could be reduced by using a chamfer instead of the circular groove. Fig. 14—Circumferential grooves have little effect on leakage in the clearance area

groove and using instead a notch with a chamfered bearing end as shown in Fig. 12. This change would give a shorter bearing without sacrificing any of the effective bearing area.

Another method of retarding the flow of lubricant along the shaft is to cut a groove around the wall of the bearing as shown in Fig. 14. As indicated in the bottom portion of this diagram, such grooves will retard the flow of lubricant in the axial direction in the high-pressure area. But, in the low-pressure area where there is clearance, the grooves will have little or no effect. Though there is less pressure to squeeze the oil out of the bearing in the low-pressure area, with a generous supply of lubricant there will always be a tendency for the oil to leak out around the shaft at this point. These circumferential grooves not only decrease the effective bearing area for a given length of bearing but also have the disadvantage of possibly forming ridges on the shaft during long periods of service.

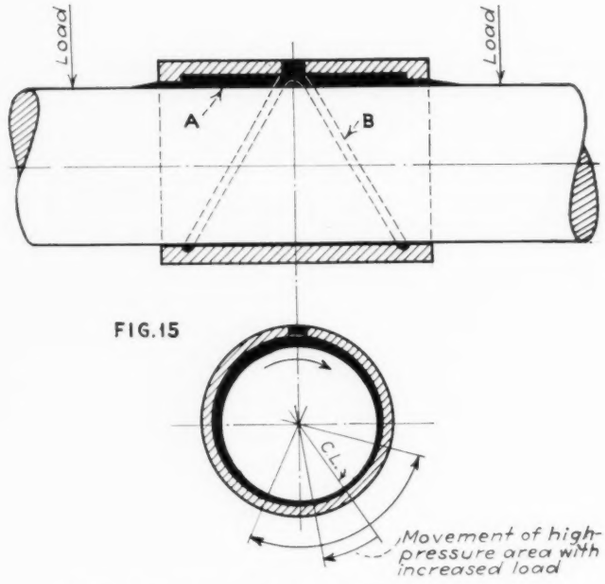


FIG. 15

Fig. 15—This method of grooving results in excessive leakage, and the grooves passing through the high-pressure area permit metal to metal contact. Fig. 16 is a better design

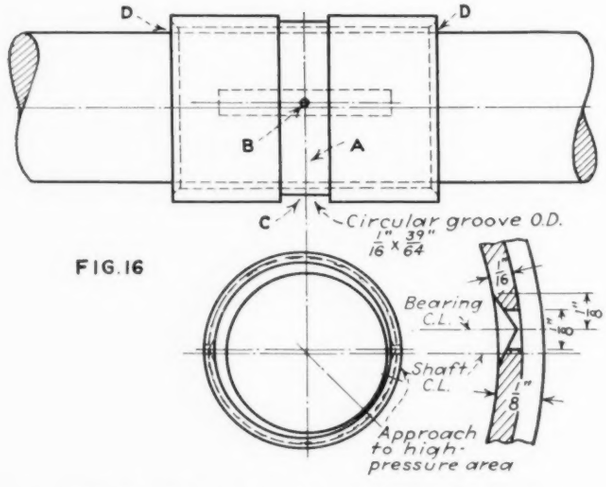


FIG. 16

Fig. 16—A groove on the outside of this bearing feeds oil through holes in the bearing wall to lateral grooves on inside of the bushing

Two different types of grooves, neither of them correct, are used in the sleeve bearing shown in Fig. 15. The longitudinal groove *A*, is cut at the top of the bearing, and the spiral groove *B* is intended to distribute the oil. Assuming the clearance area at the top of the bearing, the longitudinal groove *A* will allow too much end leakage even though it is in the low-pressure region. The spiral groove *B*, since it goes through the high-pressure area, will break up the oil film and thereby allow metal to metal contact.

A correct method of grooving this bearing is shown in Fig. 16. The inlet hole in the housing is at the top of the bearing, corresponding to the inlet hole in the bearing shown in Fig. 15. But the bearing sleeve is grooved on its outer surface. This groove carries the lubricant down the back of the bearing to the side of the wall and then through the bearing wall to the journal. At this point on the inside surface of the sleeve there is a horizontal groove on each side extending about two thirds the length of the bearing. This groove is not rectangular in cross-section, but has a V-shape, as shown in the section. Any oil fed through the hole in the bearing shell and into the horizontal groove on the approach side, will supply lubricant for the oil film. The oil that is fed on the trailing side is simply carried around by the journal and eventually deposited in the groove on the approach side. Regardless of direction of rotation of this bearing there is a plentiful supply of oil for the high pressure area. And because the grooves are not located in the region of maximum clearance, end leakage will be minimized.

The circular groove *A* need not run completely around the circumference, but can be stopped at the bottom of the oil hole *B*. However, for a solid sleeve bearing it is more economical to machine the groove completely around the bearing. With such a design in conjunction with a circulating system it might be advisable to drill a $\frac{1}{16}$ in. hole in the lower side of the bearing housing to remove the pocketed oil. The oil entry hole must be larger in area than the total cross-sectional area of the feeding hole or groove and the end clearance area. In this design, the oil hole through the housing should be $\frac{1}{4}$ in. in diameter, the hole *B* $\frac{1}{8}$ in. in diameter, and the hole in the lower side to remove the pocketed oil, $\frac{1}{16}$ in. in diameter.

When a grease-lubricated bearing requires a groove to distribute the lubricant, the grease must be applied immediately ahead of the high-pressure area and the edge of the groove must be chamfered on the high-pressure side. If the shaft is at rest for long periods of time it is advisable to place a groove parallel to the bearing axis in the high-pressure area of the bearing surface. Short helical grooves can be placed in the bearing surface to work the grease to the center.

The grease window in the bearing sleeve should extend to within $\frac{1}{2}$ in. of each end. A minimum running clearance of 0.002 in. should be allowed for shafts up to $\frac{1}{2}$ in. in diameter. This clearance must be increased to 0.005 in. for a 3 in. diameter, and 0.009 in. for a 6 in. diameter shaft.

Clearances for vertical shafts with grease lubrication can be 0.001 to 0.004 in. less than for horizontal shafts. In vertical bearings, the groove should be on an angle so that the grease will be carried upward by rotation.

Although end leakage will be reduced by the method shown here, leakage cannot be entirely eliminated without the use of separate sealing units.

QUESTION AND COMMENT

Marblette for Photo-Elastic Models

A. G. SOLAKIAN
Columbia University

● In an article in the December 1933 number, "The Scope of Photo-Elastic Analysis," the writer stated that clear Bakelite was in general use for photo-elastic tests, making use of polarized light and transparent models for stress investigations. At that time, no other available material combined the high mechanical strength and high optical sensitivity to applied stress found in Bakelite, except the Japanese material known as Phenolite. The latter material, however, could not be readily machined into intricate shapes because of its excessive brittleness. For both materials, proper annealing was required to eliminate residual stresses both in the sheet material and in the machined models. Also, the maximum size of models was limited by the size of the sheet material available.

Recently the writer found that a phenolic material, manufactured by the Marblette Corporation, has about three times the sensitivity of Bakelite. As shown by the accompanying illustration taken from the paper, "A New Photo-Elastic Material," *Mechanical Engineering*, December, 1935, the same load applied to both Bakelite and Marblette produces a greater number of fringes in the new material.

For comparing the available materials, the following table shows the stress in lb. per sq.in. required to produce one fringe per inch of thickness, and the strength in lb. per sq.in.

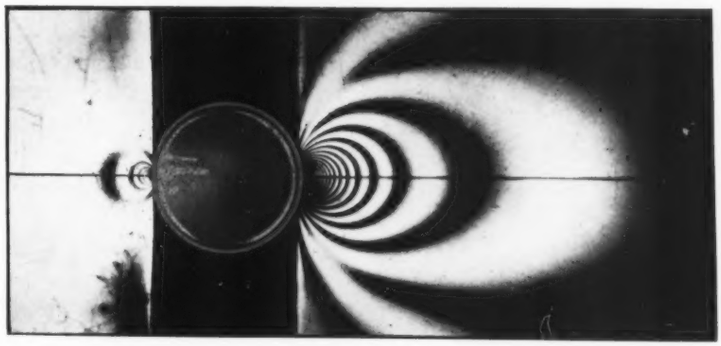
Material	*Fringe Stress*	*Elastic Limit*
Marblette	23	2,750
Phenolite	55	7,000
Bakelite	70	5,500
Celluloid	295	4,000
Glass	1,150	....

The new material has the further advantage that it can be obtained in a liquid or plastic form and cast to shape. Metal inserts can be cast in, and parts can be cemented together to form built-up models. Since the plates or models can be cast to shape, there is no restriction as to size, and no annealing is necessary for either plain or reinforced plates.

In its application to practical design problems, the new material has one advantage that opens up a whole new field of importance in mechanical design. It can be applied directly to the polished surface of an existing metal part. By a reflection type of polariscope, surface stresses can be accurately evaluated by photo-elastic methods under actual load conditions.

Because of the low cost of the Marblette material, and the ease with which it can be applied for numerous purposes, photo-elastic methods can now be adapted to a much wider range of design problems. The combination of cast Marblette models and the low-cost polarizing plates described on page 112 in this number, greatly reduces the cost of photo-elastic tests. This important design aid now can be used profitably in many routine design problems.

For photo-elastic models, Marblette (right) is much more sensitive to stress than Bakelite (left)



Effective Radius of Disk Clutches

L. D. HAGENBOOK
Design Engineer
Northwest Engineering Company

● In calculating the torque capacity of a disk clutch it has been customary to assume the pressure uniform over the entire friction surface. In terms of the inside radius r_i and the outside radius r_o, the effective radius r_e is:

$$r_e = \frac{2\,r_o^3 - r_i^3}{3\,r_o^2 - r_i^2}$$

The assumption of uniform pressure is in error, since the wear tends to be greatest at the outer surface where the velocity is greatest, and least at the inner surface of the disk. This difference in wear rate can be temporary only, as the nature of the mechanism causes wear to be uniform over the entire surface. When equilibrium is reached, the unit pressure will be least at the outside and greatest at the inside of the disk.

Recent tests on friction materials have resulted in a mathematical relation for conditions existing under equilibrium. These tests show that wear is proportional to energy absorbed, if pressure and velocity are held within the useful limits of the friction material.

Considering an elemental ring of the clutch disk, the energy absorbed or transmitted after wear has become uniform, is proportional to the areas. Inasmuch as each elemental ring has the same angular velocity, its torque dt is proportional to its area da. Since the differential area is proportional to the radius of the elemental ring, the unit pressure P in terms of a constant C is $P = C/r$.

Where f is the coefficient of friction, the elemental torque is,

$$dt = da\,p\,f\,r = 2\pi C f \int_{r_i}^{r_o} r\,dr$$

After integrating this expression, the total torque T becomes

$$T = C\,\pi\,f\,(r_o^2 - r_i^2)$$

From the same assumptions the total pressure P is given by,

$$P = 2\,C\,\pi\,(r_o - r_i)$$

Expressing the total torque T in

terms of the effective radius r_e.

$$T = Pr_e \quad f = C\pi f(r_o^2 - r_i^2)$$
$$= r_e 2C\pi (r_o - r_i) f$$

From which

$$r_e = \frac{r_o + r_i}{2}$$

After wear conditions reach equilibrium, the effective radius, therefore, is the mean radius. Besides being simple to calculate, this relation is not an approximation as has been supposed.

Flexible Coupling of Stranded Wire

To the Editor:

● I have read your very interesting article entitled, "Typical Designs of Flexible Couplings," the first part of which appeared in the January issue of *Product Engineering* and was particularly interested in the illustration and description of Fig. 4. According to the explanation, this is a coupling consisting of stranded cable, which is implied to be the same as used on dental drills and speedometer drives. This is not essentially correct as stranded cables cannot be used for this purpose. The material used is flexible shafting which is of entirely different construction and has altogether different characteristics from stranded cable. Although possibly stranded cable, which is nothing more than steel rope, might be used in very short lengths for transmitting small loads, it would not be satisfactory for heavy loads.

The power drive type of flexible shafting is used for dental drills, speedometers, tachometers, and fuel pump drives and other applications. We do not believe a stranded cable would give very satisfactory performance if used as a coupling.

—J. C. SMACK
S. S. White Dental Mfg. Company

To the Editor:

● Relative to the distinction between standard cable and flexible shafting, undoubtedly stranded wire would suffer from overstress if subjected to rotation while held at a large angular misalignment and heavy load. Regarding the suitability of using stranded cable for flexible couplings with very small misalignments, I have used this type with success in laboratory work where it would be no inconvenience at all to replace the wire if it broke. It is surprising how long a coupling like this will last. Naturally such a type of coupling would not be recommended for machine use.

—CYRIL DONALDSON
Rochester Mechanics Institute

Solution of Optical Extensometer Problem

To the Editor:

● The solution of the problem of the optical extensometer on page 469 of the December number depends on the degree of accuracy expected. It is assumed that the light source is interrupted by a slit aperture so as to impose on the mirrors a ray of light as narrow as can be dependably recorded and read on the sensitized paper drum. The thickness of the glass of the mirrors affects the solution as it will determine to some extent the separation of multiple images reflected from the second mirror (there will be either 3 or 4 of these images). Since the separation of these images also depends on the initial angle of incidence of the light ray with respect to the first mirror, no computations have been made concerning this phase of the problem. It would seem that metallic mirrors would eliminate duplicate images. The following calculations are therefore based on a single reflected image.

It is further assumed that both mirrors are initially parallel and that the first mirror is set at 70 deg. to the horizontal light beam before the specimen is loaded; also that the length of this strain is 0.001 in. The angle that both mirrors will be tipped by the 0.001 in. strain will be 0.001/0.175 cos 20 deg. = tan of angle = 0 deg. 20 min. 54 sec. tip.

The total displacement of the final image CC' is composed of two parts, the initial displacement BB' (or CD) plus the final displacement $C'D$. Assuming angle ABB' approximates 90 deg.:

$BB' = 2.484375 \times \tan\ 0°\ 41'\ 48''$
$= 0.0301$ in.

$BB' = CD$

$C'D = 1.0000 - 0.0301 = 0.9699$ in.

Assuming CB approximates DB' or X in the diagram on page 469 in the December issue, CB or X = 0.9699/tan 1 deg. 23 min. 36 sec. = 39.876 in. or $39\frac{7}{8}$ in. approximately.

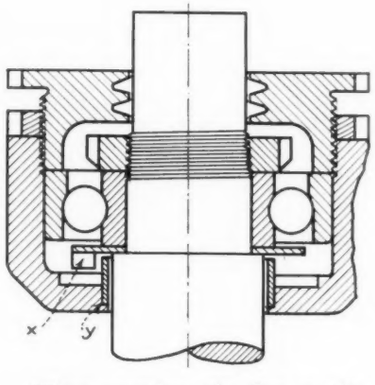
It should be noted that if the specimen is strained 0.002 in. instead of 0.001 in. as assumed in the above solution, the angle 0 deg. 20 min. 54 sec. will be more than doubled. The same disproportionate angular increase also occurs at the second mirror. The final result will not be a constant magnification of 1,000 to 1, as the magnification continually increases with further movement of the extensometer. It would therefore be necessary to compute a calibration curve with calculations for every 0.001 in. of increment.

—ROBERT W. WOOD, JR.
New Dorp, Staten Island, N. Y.

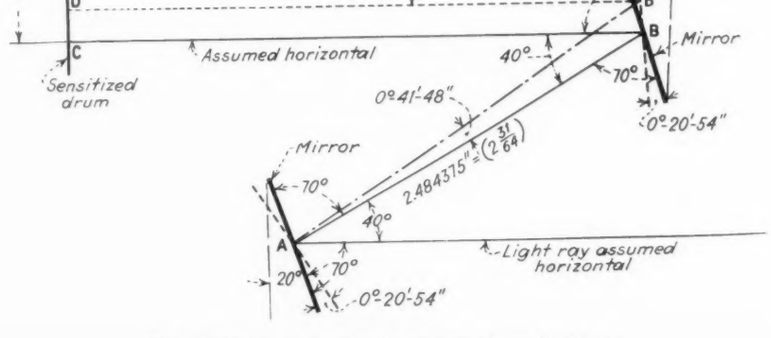
Vertical Shaft Bearing Lubrication

To the Editor:

● Some time ago I had occasion to use a simple method for automatically lubricating a ball bearing on a vertical shaft as shown in the accompanying drawing. Between the inner ring



When rotating, the flap at the periphery of the washer agitates the oil to cause splash lubrication



The length of strain is assumed as 0.001 in. to give a magnification of 1.000 in. on the drum

of the bearing and the shaft shoulder is a thin metal disk. This disk has a single slit at the periphery, forming a flap *x* when bent downward at a 45 deg. angle. A tube *y* is pressed into the housing the upper end of which determines the oil level. The washer flap, being below this oil level, scoops up the oil and splashes it into the bearing.

Recently the problem developed where two ball bearings must be lubricated, the upper bearing being approximately 6 in. above the lower bearing. The method shown is not applicable to the new design. I am wondering if some of the readers of *Product Engineering* can suggest a number of methods for lubricating these bearings. —FRED ADAMS
New York, N. Y.

[*Editor's Note:* We are sure that our readers have ideas on this subject. Any ideas submitted will be shown on these pages.]

Arguments About Bearing Pressures

● With reference to the article "Lubrication of Sleeve Bearings" by R. G. N. Evans, which appeared in December *Product Engineering*, the author states in the next to last paragraph: "Further travel of this high-pressure area is limited because it cannot cross the line of action of the load." This is not substantiated by the published reports of actual tests and I am wondering whether Mr. Evans has drawn his conclusions from actual test data or whether he is propounding a theory.
—JAMES I. CLOWER
Virginia Polytechnic Institute

To the Editor:

● Every time that I read an article on the lubrication of bearings I get all confused. Referring merely to past articles that have appeared in *Product Engineering* by Professor Bradford and the present series by R. G. N. Evans, there seems to be little agreement as to the location of the point of nearest approach between the journal and the bearing. Which is right and why? —GEORGE W. ADAMS
Detroit, Mich.

To the Editor:

● With reference to the questions raised by Professor Clower and Mr. Adams, it is true that the relative position of the journal in its bearing is shown differently in different articles. The explanation for this is that the journal will occupy different positions in the bearing depending upon the degree of lubrication, speed and loads.

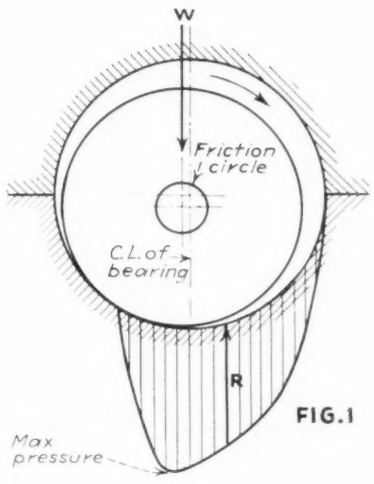


FIG. 1

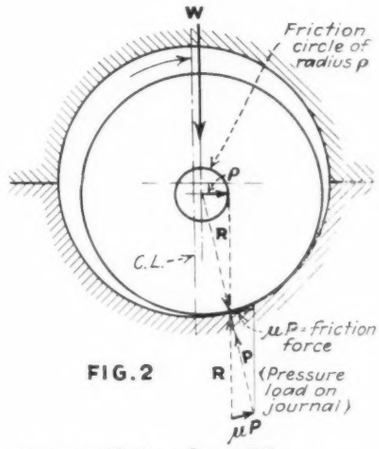


FIG. 2

For equilibrium $R\rho = \mu PR$
Also W and R are = and II
Note – From similar triangles
$\frac{R}{P} = \frac{\mu P}{\rho}$
or $R\rho = \mu PR$

When the journal starts rotating, it climbs the bearing wall as I have indicated in my articles, and shown in the accompanying Fig. 2. With grease lubrication or incomplete oil film lubrication, the journal will stay in this position indefinitely. Also, and what is highly important, it is this position that the journal takes when the oil-film breaks down and the bearing begins to score. In fact, it makes little difference to the mechanical designer what the position of the journal might be when full fluid film lubrication has been established and the film pressure displaces the journal to the position as shown in Fig. 1. The most important condition that the designer must consider is that which exists when lubrication breaks down and the bearing is likely to be ruined.

Inspections of bronze bearings that have failed in service always show that the bearing had made metal to metal contact at a point to the right of the load line, as indicated in Fig. 2. It is when the journal is in this position that the most efficient lubrication must be provided.

With reference to the question by Mr. Clower my article states "Further travel of this high-pressure area is limited because it cannot cross the line of action of the load." This statement should have been more definite. By high-pressure area was meant the center of gravity of the pressure area, that is, the resultant of all of the film pressures acting on the journal. In numerous illustrations that have appeared in my articles I have shown the high-pressure area itself extending beyond the vertical center line. However, the resultant of these pressures must be equal and opposite to the load line *W* and its position relative to the load and the center of the journal must be such that the moment of the reaction will counterbalance the frictional moment. It can readily be seen that if this reaction *R* as shown in Fig. 2 were to the left of the vertical center line, a condition of perpetual motion would exist because the moment *R* and the frictional moment would be in the same direction.
—R. G. N. EVANS
Toledo, Ohio

To the Editor:

● With reference to the problem of the position of the journal in the bearing, Fig. 2 as shown here is correct, as is also Fig. 1. The condition in Fig. 2 represents the initial stage when the journal is just starting to rotate and when no film could be established because of the absence of peripheral velocity. Of course the coefficient of friction is usually small, in the order of 7 per cent. Therefore, μ is small and radius ρ is much smaller than as show in the figure.

Actual observations on the "climbing" of the journal at the initial stage of its rotation are recorded convincingly by V. Vieweg, German experimenter in his article "Optical Measurements of the Oil Film in Bearings." This was published in the *Petroleum Zeitschrift*, 1922, page 1405. In this article, the climbing of the shaft to the right after starting, several chattering motions, and the subsequent settling of the shaft in its running position to the left are well shown photographically.

As to the forces acting, when floating on an oil film the oil film exerts hydrostatic pressure on the journal which can readily be measured by pressure gages. The pressure distribution is as shown approximately in Fig. 1. The pressure is zero near the horizontal center line and is zero near the point of nearest approach, having a maximum not far to the left of the vertical load line. The pressures are so that the resultant of all forces on the journal is vertical and equal to *W* the load, and slightly to the right of the vertical center line.
—G. B. KARELITZ
Columbia University

Typical Designs of

CYRIL DONALDSON
Rochester Mechanics Institute

The flexible elements of the couplings shown in this third installment are leather, metal, rubberized fabric, or springs

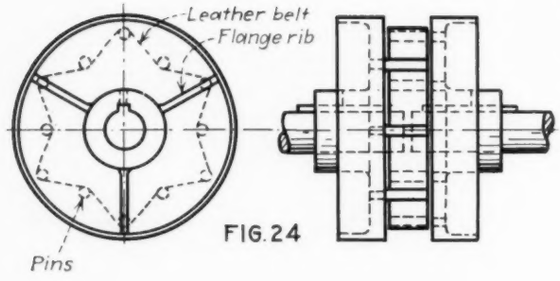


FIG. 24

Fig. 24—This Webster Manufacturing Company coupling uses a single endless leather belt instead of a series of bands, as in Fig. 22. The belt is looped over alternate pins in both flanges. Has good shock resisting properties because of belt stretch and the tendency of the pins to settle back into the loops of the belt.

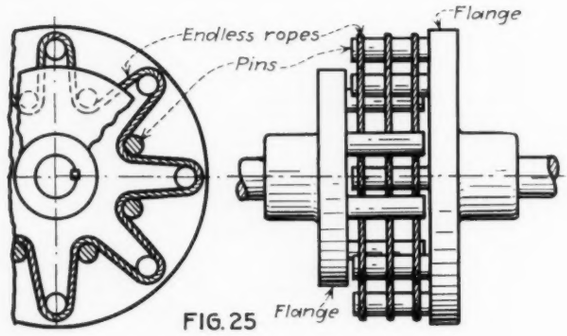


FIG. 25

Fig. 25—This coupling made by the Weller Manufacturing Company is similar to the design in Fig. 24, but instead of a leather belt uses hemp rope, made endless by splicing. The action under load is the same as in the endless belt type.

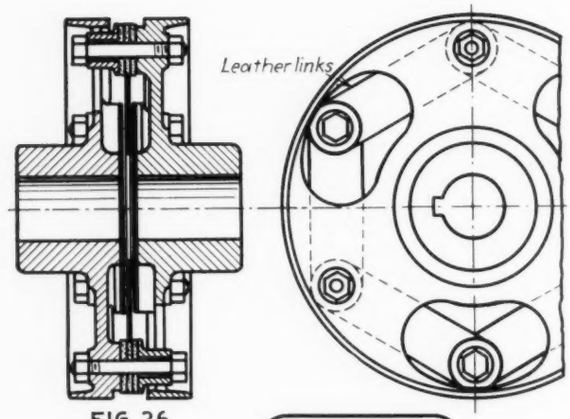


FIG. 26

Fig. 26—This Bruce-Macbeth design uses leather links instead of endless wire cables, as shown in Fig. 23. The load is transmitted from one flange to the other by direct pull of the links, which at the same time allows for the proper flexibility. Intended for permanent installations requiring a minimum of supervision.

Fig. 27—The Oldham form of coupling made by W. A. Jones Foundry and Machine Company is of the two-jaw type with a metal disk. Is used for transmitting heavy loads at low speed.

Fig. 28—The Charles Bond Company star coupling is similar to the cross type shown in Fig. 21. The star-shaped floating member is made of laminated leather. Has three jaws in each flange. Torque capacity is thus increased over the two-jaw or cross type. Coupling takes care of limited end play.

Fig. 29—Combination rubber and canvas disk is bolted to two metal spiders. Extensively used for low torques where compensation for only slight angular misalignment is required. Is quiet in operation and needs no lubrication or other attention. Offset misalignment shortens disk life.

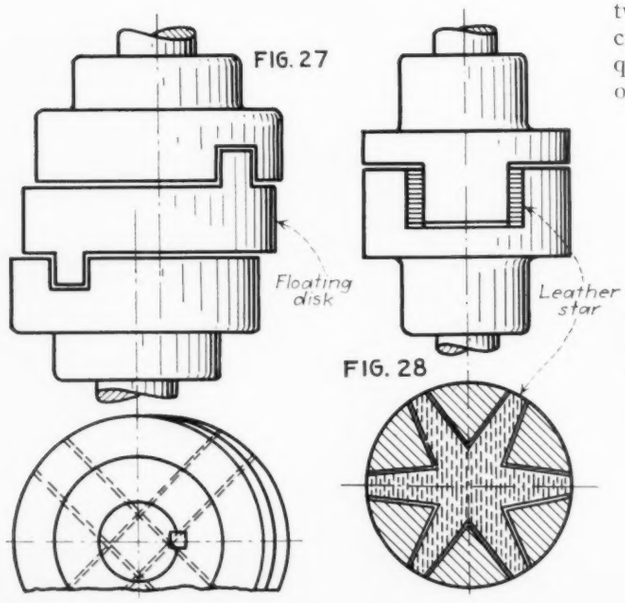


FIG. 27 FIG. 28

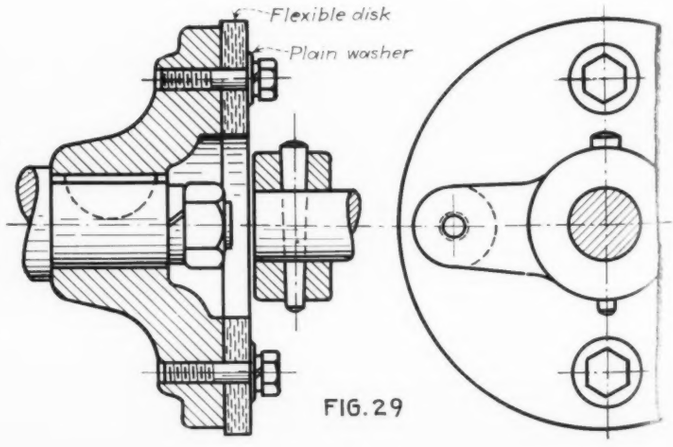


FIG. 29

Flexible Couplings–III

Fig. 30—A metal block as a floating center is used in this American Flexible Coupling Company design. Quiet operation is secured by facing the block with removable fiber strips and packing the center with grease. The coupling sets up no end thrusts, is easy to assemble and does not depend on flexible material for the driving action. Can be built in small sizes by using hardwood block without facings, for the floating member.

Fig. 31—This Westinghouse Nuttall Company coupling is an all-metal type having excellent torsional flexibility. The eight compression springs compensate for angular and offset misalignment. Allows for some free endwise float of the shafts. Will transmit high torques in either direction. No lubrication is needed.

Fig. 32—Similar to Fig. 29, but will withstand offset misalignment by addition of the extra disk. In this instance the center spider is free to float. By use of two rubber-canvas disks, as shown, coupling will withstand a considerable angular misalignment.

Fig. 33—In this Smith & Serrell coupling a flexible cross made of laminated steel strips floats between two spiders. The laminated spokes, retained by four segmental shoes, engage lugs integral with the flanges. Coupling is intended for the transmission of light loads only.

Fig. 34—This coupling made by Brown Engineering Company is useful for improvising connections between apparatus in laboratories and similar temporary installations. Compensates for misalignment in all directions. Will absorb varying degrees of torsional shocks by changing the size of the springs. Springs are retained by threaded pins engaging the coils. Overload protection is possible by the slippage or breakage of replacable springs.

Fig. 35—In another design by Brown Engineering Company, a series of laminated spokes transmit power between the two flanges without setting up end thrusts. This type allows free end play. Among other advantages are absorption of torsional shocks, has no exposed moving parts, and is well balanced at all speeds. Wearing parts are replacable and working parts are protected from dust.

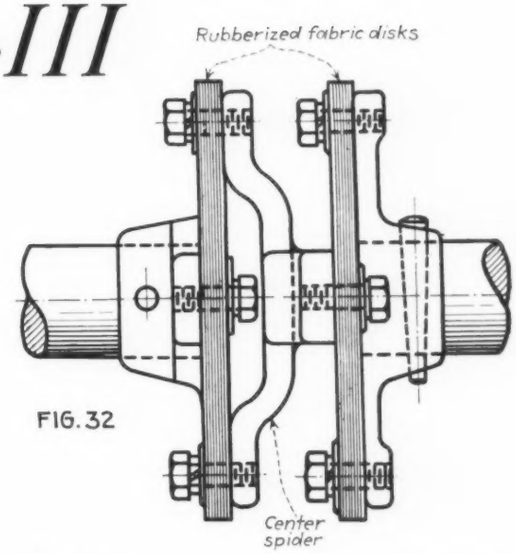


FIG. 32

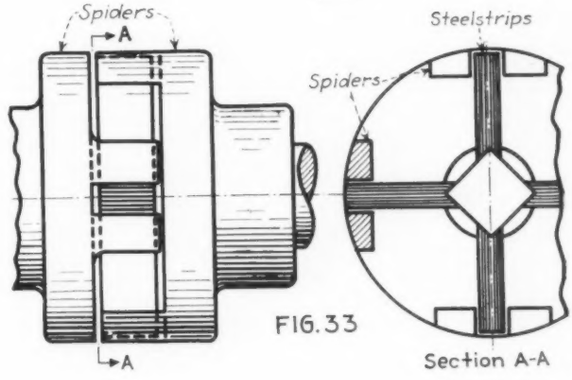


FIG. 33

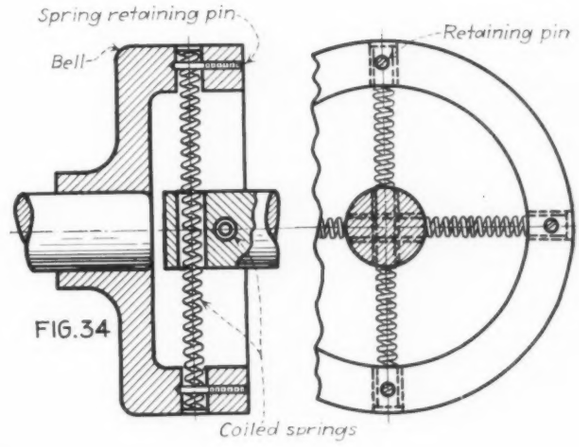


FIG. 34

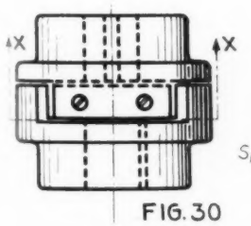
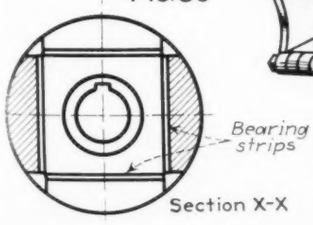


FIG. 30



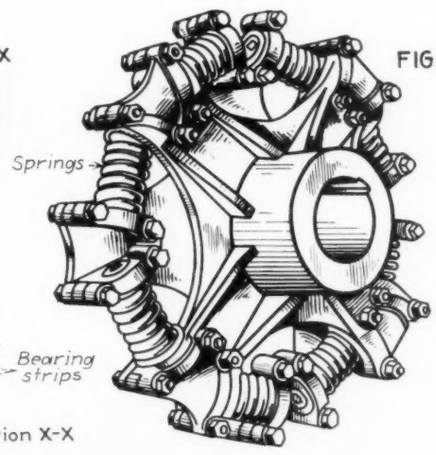


FIG. 31

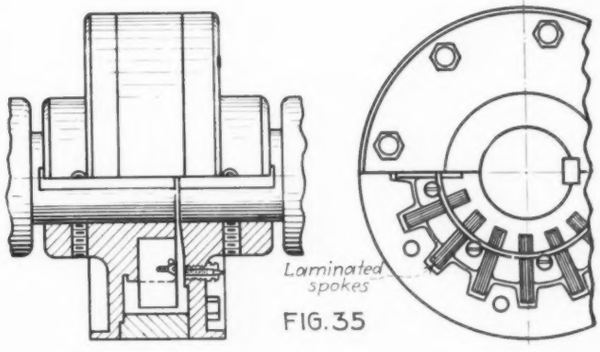
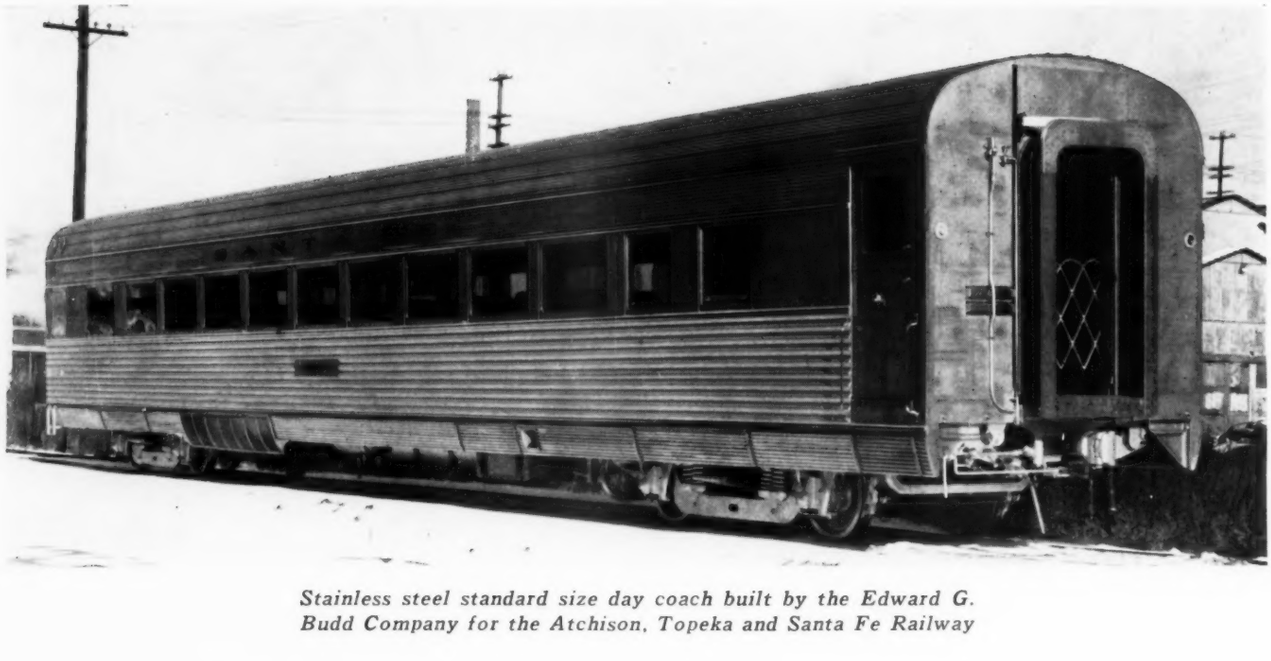


FIG. 35



Stainless steel standard size day coach built by the Edward G. Budd Company for the Atchison, Topeka and Santa Fe Railway

NEWS

Light-Weight Standard Coach Made of Stainless Steel

A STAINLESS steel railroad chair car, in which are incorporated many new light-weight structural features, has been delivered to the Atchison, Topeka and Santa Fe Railway by the Edward G. Budd Manufacturing Company.

The car, the first of its kind to be built for any railroad, is of standard size, but weighs only half as much as an ordinary Santa Fe chair car. It will be used in trains of standard steel cars.

This light-weight car is described by its builders as "the second step" in solving the railroad passenger problem. The first step, they said, was the streamlined train in which all the cars are built into a single articulated unit with a Diesel power plant in the forward car. This second step is the light-weight car designed to be coupled and which can be used in trains made up entirely of light-weight cars or mixed with conventional heavy cars and pulled by any type of power.

The car, designed for use in a long train of heavy cars, is capable of withstanding a buffing load of 400,000 lb. It weighs only 83,500 lb., and of this weight 30,000 lb. are in the two trucks. Standard passenger coaches weigh 130,000 to 150,000 lb. Because of the light weight of the car, the trucks have four wheels each, instead of the six on the ordinary heavy car.

Windows are of double panes of shatter-proof glass with nitrogen hermetically sealed between them to prevent frost and collection of moisture. To provide the utmost in riding comfort, the trucks have many special features. These include hydraulic shock absorbers, rubber insulation, and special wheel contours. Clasp brakes are actuated by unit mounted brake cylinders. Thick layers of cork and insulating blankets effectively shut out truck noise that might be transmitted through the floor.

The design of the car was developed jointly by the engineering staff of the Santa Fe and the E. G. Budd Manufacturing Company. It is the purpose of the Santa Fe to operate the car in regular service in its heavy main line trains to develop all the facts as to the serviceability of this type of car.

New Polarizing Glass Opens Way For Many New Design Possibilities

Color movies in three dimensions, glare-less headlights, full scale photo-elastic test models, now obtainable with laminated clear glass

UNTIL recently, many practical uses for polarized light have been restricted because of the limitations in the size of polarizers. About the only convenient polarizer was a Nicol prism, and the largest ones in general use gave a beam only about $\frac{3}{4}$ in. in diameter. Made of specially prepared crystals of Iceland spar, such polarizers are expensive and hence limited in use. The recently announced new material called Polaroid, developed by the Land-Wheelright Laboratories makes available polarizing "glass" that is relatively inexpensive and obtainable in sheets as large as 14 in. wide and several feet long. For polarized beams of the same area, Polaroid costs about one-fiftieth as much as a Nicol prism.

Polaroid is nothing more than two sheets of glass between which a thin sheet of cellulose acetate material is cemented. The only fundamental difference between this glass and other laminated glass is that the cellulose

sheet, only a few thousandth of an inch thick, has suspended in it minute artificial crystals each with its polarizing axis in the same plane. The result is that only the components of the transverse light vibrations in one plane are transmitted and the vibrations in the other planes are blocked.

For three-dimensioned moving pictures the films are taken with the ordinary stereoptican machine. The two films are then projected on the screen as superimposed images. But a plate of Polaroid in each projector polarizes the one image in one plane and the other image in a plane perpendicular to the first. When the images are seen through special spectacles of polarized glass, each eye sees but one of the images, and the separate images formed in each eye thus give the three-dimension illusion.

Previously the same effect was obtained by using two films of different color and viewing the superimposed pictures through spectacles having color filters. Natural color moving pictures were not possible with this system.

Another use of Polaroid is in glareless automobile headlights. With all car headlamps and windshields equipped with the glass polarized 45 deg. to the horizontal, all in the same plane with respect to the car, approaching cars would have their headlights and windshields polarized at an angle of 90 deg. to each other. To each driver the approaching headlights would appear dim, but the illumination created by his own headlights would not be affected.

For photo-elastic analysis an ordinary lamp bulb and two sheets of polarizing glass is all that is needed. The stressed model is placed between the two polarizing sheets with their axis set at an angle of 90 deg., and the stress fringes observed or photographed in the ordinary manner. Thus full size models a foot high and several feet in length can be tested readily. For models with a maximum dimension of 10 in., a complete analyzer can be built for $35 or less.

By placing a sheet of clear, colorless cellophane between two sheets of Polaroid, brilliant colors appear, each color changing to its completement as one of the polarized plates is rotated. Unusual stage effects and other commercial applications of this phenomena undoubtedly will be found.

Glaring reflections from painted or bright metal surfaces consist of a large proportion of scattered light. Such surfaces can be examined or inspected with greater accuracy when viewed through sheets of Polaroid, or through spectacles fitted with polarizing lenses which stop the glare but not the general illumination.

In the first demonstration of Polaroid, Edwin H. Land, who invented the material, showed actual examples of its applications, including glareless headlights, photo-elastic analysis, one-way vision for windows, glareless inspection of painted surfaces, and three-dimensional still pictures and movies.

MEETINGS

American Gear Manufacturers Association—Annual meeting, Adelphia Hotel, Philadelphia, Pa., April 20-21. J. C. McQuiston, secretary, Penn-Lincoln Hotel, Wilkinsburg, Pa.

American Society for Testing Materials—Annual meeting, Atlantic City, N. J., June 29-July 3. R. E. Hess, assistant secretary, 260 S. Broad St., Philadelphia, Pa.

EXHIBITIONS

Packaging Exposition — Hotel Pennsylvania, New York, N. Y., March 3-6. American Management Association, 232 Madison Ave., New York, N. Y.

National Oil Burner Show—Convention Hall, Detroit, Mich. April 14-18. C. F. Curtin, 30 Rockefeller Plaza, New York, N. Y.

New Molding Company Organized

EFFECTIVE February 1, business of the Molded Products Division of the American Record Corporation of Bridgeport, Conn., and Scranton, Pa., was taken over by a new company, The Consolidated Molded Products Corporation. Operations are now consolidated at Scranton with general offices at 309 Cherry Street.

The molding plant taken over by this new organization has been in continuous operation for more than half a century, starting originally in the manufacture of buttons. Their service today comprises molding in all plastic materials for all industries.

Joint Meeting on Cast Metals

FOR the second time, the Chicago Chapter of A.F.A., the Chicago section of A.S.M.E., and the Western Society of Engineers are cooperating to hold a joint meeting to present engineering data on cast metals. The meeting will be held in the Auditorium of the Engineering Building, 205 West Wacker Drive, Chicago, on March 16. George M. Zabel, foundry superintendent, Fairbanks Morse & Company, Beloit, Wis., will speak on "Engineering in its Relation to the Foundry," and G. P. Phillips, metallurgist, International Harvester Company, Chicago, on the "Selection and Use of Cast Metals for Machine Parts."

Mr. Zabel's talk will stress the importance of designing castings for production and will show how changes in design have made possible more economical production and an improvement in the physical properties of the castings. Mr. Phillips will deal with the factors that must be taken into consideration when selecting materials for castings.

An exhibit planned in connection with the meeting will illustrate the development of materials possessing superior mechanical properties which are now available as contrasted with those available a few years back.

Do You Know That—

- Reduction in area for 5 per cent chromium steels is increased from 12 per cent to more that 60 per cent just by the addition of a small amount of columbium.

- A new permanent magnet material is hardened not by quenching but by annealing, and will support 60 times its own weight.

- Seamless nickel tubing is now available as small as 0.005 in. in diameter with a wall thickness of 0.0008 in. It weighs only 0.0000406 lb. per ft.

- Many vagaries in instrument performance can be traced to impurities in the bore of capillary tubing. One company uses continuous hydrogen annealing furnaces in the manufacture of the tubing in order to remove such impurities.

- By a new process the amount of glass used in a ketchup bottle can be made into a single strand long enough to reach from New York to Los Angeles. The fibers are only 0.00025 in. in diameter.

- Paint holds much better on anodized aluminum sheets if the sheet is sprayed with a primer as soon as it has dried after leaving the anodizing bath.

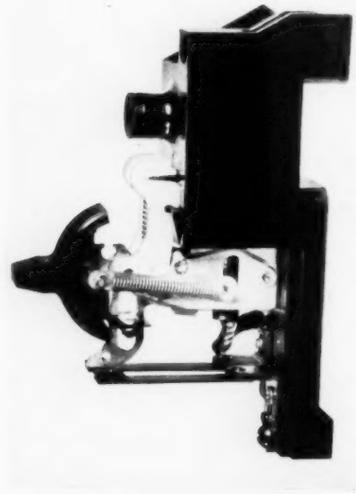
- Synthetic rubber not only resists heat, gasoline and oils better than the natural material, but is now being used with newer types of refrigerants that attack rubber rapidly.

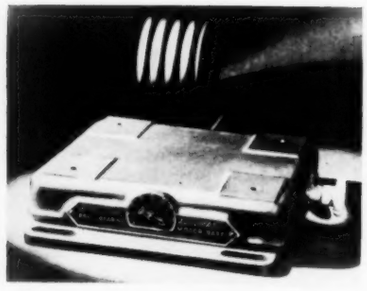
For more details write the Editor

New Materials and Parts

Air Circuit Breaker

Redesigned breaker replaces fuses and is used in the control and protection of industrial circuits, electric refrigerators and ranges and other equipment. A "mid-position" of the handle is provided to indicate automatic tripping and is between the on and off positions. New features incorporated are easier operation, a mechanical "pry" to open the contacts when manually operated, and in double and triple-pole units, a latch that is more resistant to shocks. Available for 230-volt a.c., 125 to 250-volt d.c., 50-amp. use. General Electric Co., Schenectady, N. Y.



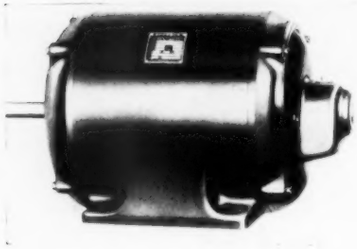


Automatic Motor Base

Is used in conjunction with the "Vari-Pitch" V-belt sheave to maintain a uniform belt tension throughout the speed range adjustment. A dial indicator shows the belt tension. The upper half of the "Straitline" base on which the motor is mounted, is supported by four totally inclosed ball bearings, lubricated for the life of the unit. The handwheel on the base controls the speed of the sheave and simultaneously moves the motor in a straight line, a sufficient amount for proper belt tension. Allis-Chalmers Mfg. Co., Milwaukee, Wis.

Thermostatic Control

For bonnet, fan and duct service has adjustable range and adjustable differential. The mounting bracket is swiveled for convenience in mounting. When applied with the element in the plenum chamber of the furnace, the thermostat may be used as a high temperature limit control or as a fan switch to permit the fan to operate only when heat is available. Has a voltage range to 250 volts. Spencer Thermostat Co., Attleboro, Mass.



Single-Phase Motor

Type SR motor has an armature with both a squirrel-cage winding and a repulsion or commuted winding. Both windings operate as such at all times; the commutator of the repulsion winding is not short circuited as the motor attains speed and the squirrel-cage is not opened at slow or starting speeds. The starting action is said to be smooth and steady comparable to that of a polyphase motor; also that there is no heavy sparking with resultant commutator pitting. Used for air compressors, commercial refrigerators, pumps, grinders, choppers, mixers and other devices having a heavy starting load. The Emerson Electric Mfg. Co., St. Louis, Mo.

Capacitor-Motor Controller

Is a transformer speed regulator for high and low-torque capacitors, with and without full-voltage starting for single-phase and polyphase fan motors. Unit consists of a tapped auto-transformer and snap switch, and for providing full-voltage starting, a relay affording automatic transfer to the desired running position. Snap switch provides for three positions, and other low speeds can be obtained through reconnection of transformer taps to the snap switch, giving a total of seven reduced speeds. Regulators are for single-phase motors to 2 hp. and for polyphase motors to 5 hp. Regulator is reversible or can be operated from external sources such as thermostats, with special additions. General Electric Co., Schenectady, N. Y.

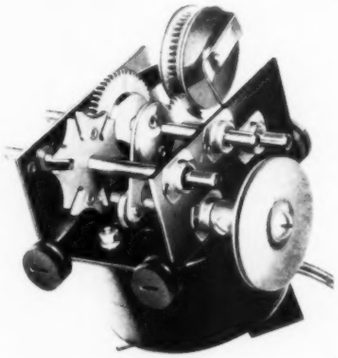
Mycalex Insulation

For applications requiring a non-warping, high-strength di-electric, a composition material of powdered mica and glass is molded under pressure and heat. Molded parts are said to be unaffected by moisture and temperatures up to 800 deg. F. Can be molded and machined to close tolerances; is readily drilled and tapped, and ground after molding. As compared with other insulating materials, Mycalex is said to have better electrical properties, lower

losses at radio frequencies, higher mechanical strength and lower surface leakage. Available in sheet and rod as well as molded parts. Electronic Mechanics, Inc., 201 E. 12th St., New York, N. Y.

A.C. Motor with Gear Reduction

Small gear reduced motor weighing less than 3½ lb. having 70 in. lb. torque and four ¼ in. diam. cross shafts. Has Geneva motion for six-station index. Has a cooling fan for air circulation said to make operation possible under extreme heat conditions. Bearings are of self-lubricating type. High-speed shaft rotates 56 r.p.m. and slow-speed shaft rotates 6 r.p.m. Is said it will not burn out if stalled on full line voltage. Model 85 runs on a.c. 110 volts. A. G. Redmond Co., Flint, Mich.

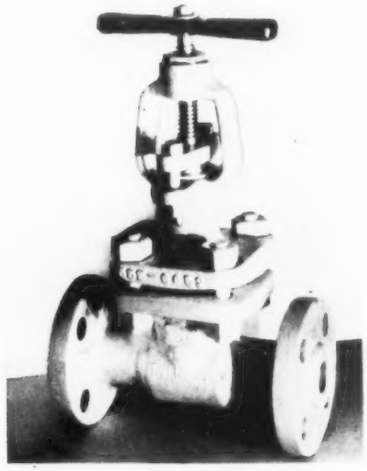


"Protectit"

Combination control device which functions as a manually operated switch with automatic current overload protection. A bimetal strip, serving as a mechanical latch, connects the switching member and its movable contacts with the handle. Overheating of the strip by passage of an electrical overload through it, causes it to bend sufficiently to break the circuit. Circuit rupturing is "Dualbreak." On and off manual switching is positive toggle make and quick break. Adaptable for wall or appliance mounting. Molded case is 3⅜ x 1½ x 1½ in. Type BH makes use of 19 interchangeable heater elements ranging from 1.3 amp. to 14.2 amp. Type B with 10 fixed ratings range from 1.3 amp. to 15.0 amp. Maximum carrying capacities average 10 per cent greater than amp. ratings. Both types have a maximum voltage rating of 250 volts, a.c. or d.c. in any amp. with a maximum of 1 hp. rating. Colt's Patent Fire Arms Mfg. Co., Hartford, Conn.

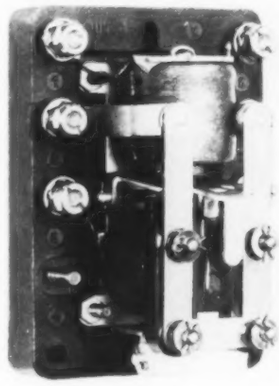
High-Pressure Valve

Forged steel globe valve for pressure ratings to 1,500 lb. has tongue-and-groove bolted bonnet. Used for high temperature service. Available either in cone or plug type or with valve seat ring and disk made of heat-treated stainless steel. All wearing parts are renewable and are said to be readily accessible. It is claimed that wire drawing, indenting and galling is not possible. Hancock Valve Div., Consolidated Ashcroft Hancock Co., Inc., Bridgeport, Conn.



Electrical Reset Relay With Mechanical Latch

For three and four poles with single or double throw. Relays are provided with two coils, one for opening and one for closing. Energizing the closing coil picks up the armature which is then held closed by a metal latch. The latch engages a bakelite keeper on the armature. This combination is said to eliminate sticking and to reduce wear. Energizing the release coil trips the latch and allows armature to drop out. Can be used for thermostatic control on electric ranges and electric heating equipment or in applications where voltage is subject to wide fluctuations. Struthers Dunn, Inc., 139 North Juniper St., Philadelphia, Pa.



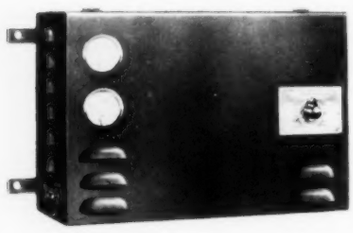
"Sure Flow" Pump

Direct motor-driven pump for coolants, water, oil or fluids charged with abrasive. Said to maintain its prime without submerging any part of pump in the fluid. Supplied with horizontal or vertical base for upright or wall mounting. Has a capacity of 4 gal. per min. Motors of ⅛ hp. rating are used for low pressure and ⅓ hp. motors for high pressure. Pump is adaptable where a limited amount of coolant is required such as lathes or other single spindle machine tools. Other sizes to 100 gal. per min., are available. Acme Machine Products Co., Inc., Muncie, Ind.

Controlled Rectifier

Used for supplying d.c. output from a commercially constant, single-phase, 110 volt, a.c. line. Consists of Hysterset, rectifier and filter units assembled in a metal case arranged for wall mounting. The Hysterset func-

tions to hold the d.c. output voltage within plus or minus two per cent from approximately 1/10 to full load. On sizes covering a range from 30 to 250 watts, the efficiency is said to be greater than 50 per cent and the power factor greater than 65 per cent. Ward Leonard Electric Co., Mount Vernon, N. Y.



Seamless Steel Tubing

Is used for service lines in industrial and commercial refrigeration and air conditioning units and for oil and feed lines in hydraulic circuits for machine tools. Said to have a ductility approaching that of copper and to bend readily. Can be flared with hand tools to accommodate standard S.A.E. fittings. Is made of a special low carbon steel that is resistant to corrosion and free from oxide or scale. Made in outside diameters equivalent to copper tubing. Illustration shows $\frac{1}{4}$, 5/16 and $\frac{1}{2}$ in. outside diameter tubing flared with hand tools. Summerill Tubing Co., Bridgeport, Pa.

Small Drawing Board

Is made of pressed wood having four raised side and end guides against which a triangle slides. The triangle takes the place of a tee-square. No thumb tacks are used as the paper is fastened by a spring clamp at each corner. The clamps are so shaped to elevate the corner of the triangle over the adjacent side guide permitting the triangle to pass freely beyond the limits of the board. Therefore there are no locations on the board that cannot be reached with an 8 in. triangle. The paper clamps are lifted by finger pressure against plugs at the reverse side of the board. The surface against which the drawing is made is hard enough to resist indentation of the hardest pencil. There is no wood grain to cause the pencil to wander. Available in several sizes, the smallest of which accommodates letter size sheets of $8\frac{1}{2}$ x 11 in. This size weighs less than 1 lb. and can be carried in a briefcase. H. E. Twomley, 7154 Magnolia Ave., Riverside, Calif.

Anti-Friction Molding Material

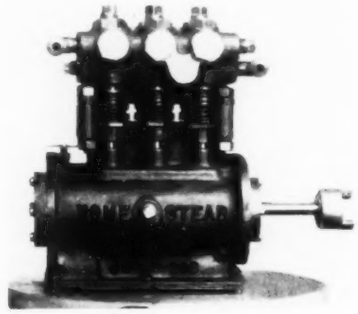
Durez 1564 contains 10 per cent graphite which is claimed to give this phenolic molding compound 40 per cent greater impact strength. Is used for applications where constant friction plus impacts of varying degrees require material of greater resistance. Cams, automobile door bumper shoes, hardware parts, refrigerator latch bolts or machine parts are examples to which this molding compound can be applied. General Plastics, Inc., North Tonawanda, N. Y.

Hydraulic Control Panel

For mounting on machine to provide cycles of operation such as rapid approach, feed, adjustable dwell and rapid return. A second or slower feed may be obtained. The operation of feed and rapid traverse sequence is controlled by adjustable dogs. The control panel consists of pilot, reversing and selector valves, all of which are operated from a low-pressure source. There is also a relief valve for both high and low pressure circuits. The hydraulic circuit is said to be self-locking and to provide smooth uniform feed and shockless high speed traverses. "Climb" cutting on milling machines can be accomplished with the use of this panel. Sundstrand Machine Tool Co., Rockford, Ill.

Positive Displacement Pump

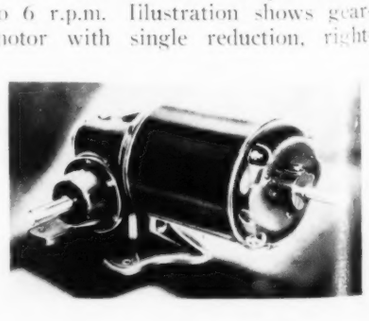
Small plunger reciprocating pump to fill the gap which exists between mechanical lubricator type and rotary centrifugal type of pump. Has three single acting plungers of $\frac{3}{4}$ in. diam.



and $\frac{5}{8}$ in. stroke. Poppet type inlet valves are of monel metal. Outlet check valves are of ball type with adjustable limited lift for high-speed operation. Cylinder head is made of corrosion-resisting iron alloy. Depending on speed, pumping capacity is variable from 15 to 120 gal. per hr. Has a pressure range of 500 lb. requiring $\frac{1}{6}$ to $\frac{1}{2}$ hp. driving power. Can pump water containing various percentages of alkaline compounds, fuel or mineral oils, and a number of chemicals. Is used for boiler feed, boiler compound injection, condensate returns, small hydraulic presses, oil burners, or air-conditioning apparatus. Homestead Valve Mfg. Co., Coraopolis, Pa.

Gear-Motor

Fractional horsepower gear-motor for direct connection to stokers, agitators, conveyors, low-speed pumps, mixers and other applications requiring special speeds. Available in single or double-reduction types with right-angle shaft drive; and in single, double and triple-reduction types with parallel shaft drives. Furnished at speeds down to 6 r.p.m. Illustration shows gear-motor with single reduction, right-

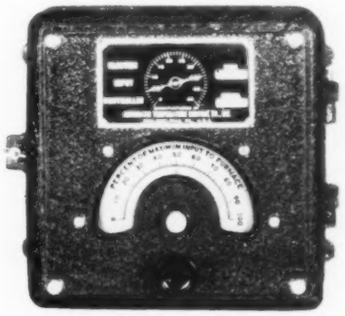


angle drive with motor of ⅛ hp. capacity, repulsion-start-induction type with special high-speed shaft extension. Motor shaft is ball-bearing mounted. Low-speed output shaft supported on two roller bearings. Worm gear is of phosphor bronze and worm of heat-treated nickel alloy steel. Wagner Electric Corp., St. Louis, Mo.

Current Input Controller

For furnaces, ovens, platens and other electrically heated units. Model M is used to prevent a constant load from rising in temperature beyond a desired and readily adjusted setting. Model MT is the same as M and is additionally provided with built-in time control to bring input-control into action after an adjustable period. Models F and FL are for control where loads are variable, automatically adjustable, the former having a slight lag in furnace response, the latter serving where lag is greater.

The external knob adjusts position of 10-amp. snap-switch in relation to a rotary cam. For capacities greater than 10 amp. at 110 volts a.c., mercury contact tubes rated to 90 amp., 110 volts a.c., are available. The mercury tubes are solenoid operated with spring cushioning to prevent undue mercury agitation. Unit is housed in hinged and gasketed aluminum case 8¼ in. high, 9¾ in. wide and 4¾ in. deep. Automatic Temperature Control Co., Inc., 34 East Logan St., Philadelphia, Pa.



Pencils for Blueprints

Two new pencils, silver and gold, with "Verithin leads." The metallic tints provide a sharp contrast to the dark background of blueprints or other colored paper. Can be erased easily. Used to designate flashings, piping, wiring, and facings of brass, copper, steel, tin, or other metals. The hexagonal wood casing contains "leads" of the same size as ordinary pencils. Eagle Pencil Co., Inc., 703 E. 13th St., New York, N. Y.

MANUFACTURERS' PUBLICATIONS

Air-Operated Controllers — The Brown Instrument Co., Wayne and Roberts Ave., Philadelphia, Pa. Folder, 8 pages, 8½x11. Shows constructional features and use of Air-o-Line controllers having a throttling range and automatic reset dials. Motor valves and valve positioners are also explained in detail.

Ball Bearings—The Bearings Company of America, Lancaster, Pa. Data book contains dimensions, load ratings and other information for all sizes and types of ball bearings. Has simplified tables for easy interpolation of loads at in-between or odd speeds. Descriptions of pre-lubrication and inclosure features are also given. Book is wire bound and lays flat when open.

Bulk Materials Handling Machinery —Palmer-Bee Co., Section 106 of General Catalog 100, 288 pages, 7¼x10¼ in. General conveying machinery is illustrated with photographs and dimensioned line drawings. Includes buckets, chains, gates for conveyors and bins, shaker, revolving and rotating screens, clutches, bearings and truck wheels.

Fiber, Spauldite and Armite — Spaulding Fibre Co., Inc., 310 Wheeler St., Tonawanda, N. Y. Engineering Data Book. 19 pages, 8½x11 in. Gives electrical, mechanical and chemical properties and presents data relating to the workability and behavior of hard vulcanized fiber, Spauldite (laminated phenolic) and Armite (fish paper). Articles made of rods, sheets and tubes are shown. Fabricating characteristics are discussed.

Friction Material—The Manhattan Rubber Mfg. Division of Raybestos-Manhattan, Inc., Passaic, N. J. Catalog, 11 pages, 8½x11 in. Technical descriptive data on nine types of friction materials for industrial applications. Used as brake or clutch lining on cranes, dredges, power shovels, excavators, grab buckets, hoists, elevators and gas engines. Methods of application are shown with line drawings.

Machinery and Tools — Brown & Sharpe Mfg. Co., Providence, R. I. Catalog No. 140, 672 pages, 4¾x6¾ in. General catalog of machines, tools, cutters and hobs, arbors, collets and adapters, screw machine tools and various types of pumps.

Methods for the Fabrication of Lukens Nickel-Clad Steel — Lukens Steel Co., Coatesville, Pa. Bulletin T-4, 12 pages, 8½x11 in. Written as an article, the text contains information on the construction, materials and uses of clad-steel. Physical properties are given in tabular form. Hot and cold working, and various types of welding are discussed. Design of welding joints are shown in line drawings. A list of more than 80 industrial applications is included. Dimensions are given for plates of $\frac{3}{16}$ to 1 in. thick in rectangular and circular shapes.

"One-Ten Thousandth of an Inch" —The Dumore Co., Racine, Wis. Booklet, 22 pages, 4½x6¼ in. A summary of the processes of gaging shows a number of instruments used in the manufacture of interchangeable parts. Classification and kinds of fits and tolerances are tabulated and discussed. An interesting booklet from which the engineer can obtain many pointers on the subject of gaging refinements.

Pillow Blocks — SKF Industries, Inc., Front St. & Erie Ave., Philadelphia, Pa. Booklet, 8 pages, 8½x11 in. Pillow blocks equipped with spherical roller bearings of the self-aligning type for use as railroad journals, on rollnecks, crusher shafts and other heavy duty applications. Blocks are of split construction with felt sealing rings.

Properties of Toncan Iron—Republic Steel Corp., Massillon, Ohio. Booklet, 12 pages, 3½x6¼ in. Condensed and recent information on rust resisting copper-molybdenum iron. Physical properties and constants are listed and features of the material are discussed. Forms in which the iron is available are also given.

Relays—Struthers Dunn, Inc., 139 North Juniper St., Philadelphia, Pa. Catalog C, 28 pages 8½x11 in. Lists various types of relays, including industrial control and power transfer type for heating, lighting and motors. Also midget relays in single, double and multiple arrangements. Mercury, mechanical latch-in with electrical reset, instrument controlled and many other relays are included. Various types of thermostats are shown. Illustrated with photographs, wiring diagrams and dimensioned line drawings.

Sheaves for Texrope Drives—Allis-Chalmers Mfg. Co., Milwaukee, Wis. Bulletin 2188-A, 16 pages, 8½x11 in. Revised list includes "Duro Brace Texsteel" sheaves for stock drives to 15 hp. Dimension tables of all sheaves, hp. and center distances for all recommended stock ratios are also given.

Temperature Determining Device — Ideal Commutator Dresser Co., Sycamore, Ill. Folder, 4 pages, 8½x11 in. Discusses the use of "Tel-Temp" gages for motors, bearings and other machinery. Shows how the device is applied to open and closed motors, generators, bus-bars, pump bearings and condensate pumps.

Twenty-five Years of Achievement—The Porcelain Enamel & Mfg. Co., Baltimore, Md., 16 pages, 11x14 in. Anniversary pictorial review of the manufacture, control and testing of "Pemco" porcelain enamel. Ground coats, cover coats and flux enamels for sheet iron and cast iron are discussed.

BOOKS AND BULLETINS

The Metal-Iron Alloys of Iron Research, Monograph Series

H. E. Cleaves and J. G. Thompson. 574 pages, 6x9 in. 113 illustrations, 95 tables. Blue clothboard covers. Published by the McGraw-Hill Book Co., Inc., 330 West 42d St., New York, N. Y. Price $6.

This monograph has been prepared at the National Bureau of Standards of the United States Department of Commerce as part of the Bureau's contribution to Alloys of Iron Research and has been approved by the Bureau's Director. A first edition, the book is a review of available information on the preparation and properties of high-purity iron. In Part I, containing three chapters, laboratory and commercial production of electrolytic iron is discussed with an author's summary at the close of each chapter.

The nine chapters in Part II contain concise but comprehensive and essential information of the structure and the thermal, electric, magnetic, chemical and mechanical properties of high-purity iron. Heat-treating and aging are also discussed. A 70-page bibliography, and a name and subject index complete the book.

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Diesel Engineering Handbook

L. H. Morrison. 920 pages, 6x9 in. More than 800 illustrations. Red leatherette covers. Published by Diesel Publications, Inc., 192 Lexington Ave., New York, N. Y. Price $5.

"A most comprehensive and authoritative treatise ever published on diesel engines." So says Edgar J. Kates, consulting diesel engineer of New York City. Mr. Kates further says, in part, that the text presents the "whys and wherefores" of this type of power.

In the first of 40 chapters, the historical development of the internal combustion engine is shown. The basic diesel principles are then brought out in Chapter II by comparison with the common gasoline engine. Two chapters following in logical sequence are on fuel-injection sytems and combustion systems with a discussion of the principles of these parts. Separate chapters deal with slow, medium and high speed designs. A 32-page section on engineering fundamentals follows giving diesel terminology formulas, charts and tables.

A series of chapters takes up all the important components of diesel engines from frames and cylinders to fuel pumps and governors. In each instance design features are discussed. These sections are profusely illustrated with cross-section line drawings and photographs. A chapter on combustion and fuels defines the general engineering terms, briefly reviews the results of research, and concludes with a table characterizing the methods of fuel-air mixing used in practically all modern American makes of diesel engines.

Description of new equipment and records of recent tests are included for diesel-powered trucks, tractors, railtrains, locomotives and aircraft.

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Technical Aerodynamics

Karl D. Wood. 330 pages, 6x9 in. 300 illustrations, 41 tables. Blue clothboard covers. Published by McGraw-Hill Book Co., Inc., 330 West 42d St., New York, N. Y. Price $3.50.

The text of this first edition has been used in mimeographed form as a textbook for college courses in aerodynamics. The book emphasizes fundamental principles of mechanics and hydrodynamics rather than technical details of construction and operation. The book is divided into 11 chapters and 8 appendixes. At the end of each chapter are given problems to which answers are supplied.

Among the chapters are discussed properties of air, airfoil characteristics, parasite drag, engine and propeller characteristics, airplane performance and performance estimation. Other sections deal with longitudinal, lateral and directional stability, special flight problems, model testing and flight testing, and hull and float characteristics for seaplanes. The appendixes contain all the tables, charts, diagrams, formulas, figures and photographs necessary to the text. Four pages of notations give the various symbols and their definitions as used in the study of aerodynamics.

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A Rubber Plantation in New Jersey

Thiokol Corporation, Yardville, N. J. About 20 pages, 5x7½ in. Clothboard covered.

An entertainingly written, quick-reading book, giving the story of the discovery of Thiokol by Dr. J. C. Patrick, telling how it is manufactured, its properties and typical applications. Written entirely in "story form," and pleasingly illustrated with black and white drawings by John O'Hara Cosgrove II, the book is not only informative, but also easy to read, much in a humorous vein.

After briefly describing the discovery of this synthetic rubber and its method of manufacture, the book tells about its application and use for washers and gaskets that must be oil or gasoline resistant, its use in automobiles, airplanes and the oil industry for hose, tubing, valves and similar applications, press rollers and blankets, printing plates in the printing industry and Thiokol substituted for lead sheathing for electric cables.

The closing paragraph is typical of the style of presentation. "In fact you may now be able to imagine 'Thiokol's' nack of doing a lot of things and the way it benefits your industry. Very possibly this rambling history does not mention your industry. No matter. Synthetic rubber is a vital force in swaddling clothes. . . . you'll meet it face to face one day. When you do, look us up!"

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Stainless Steel

Joseph T. Ryerson & Son, Inc., 16 Lock Box U, Chicago, Ill. Booklet, 8 pages 11x14 in.

A deluxe photographic presentation of the uses of Allegheny stainless steel in all lines of industry. Of particular interest is a natural color photograph 14x22 in. in size showing household utensils and appliances, furniture, machine parts, jewelry and automobile headlights. Other products illustrated, made of this metal are chains, springs, screening, bolts, screws, rivets, and wire. The further use of stainless steel plates, sheets and strip stock is shown. This 7-color booklet is one of the most beautiful presentations of fabricated metal parts we have seen. Available to those using or contemplating the use of stainless steel by request on company letterhead.

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Ball and Roller Bearing Engineering Manual

The Fafnir Bearing Co., New Britain, Conn. Manual No. 35, 197 wire-bound pages, 8½x11 in. Cardboard covers.

A comprehensive review of the principles affecting the selection, application and operation of ball and roller bearings. Of interest to engineers is an up-to-date résumé of the engineering fundamentals involved in correlating design requirements, load characteristics and service conditions to the choice of the particular size and type of bearing. Vector diagrams and load formulas for belt drives, chain and rope drives and spur, bevel, helical and worm gears show the forces acting on the bearings.

The book is divided into five sections for easy reference. Section I reviews the history and development of anti-friction bearings. In Section II, 30 pages are devoted to a broad analysis of bearing selection for type, size and life expectation. In the 118 pages of Section III, detailed dimensions, load ratings and functional characteristics are given. In the fourth section, fits and fitting methods are discussed. Housing and mounting instructions and advice on lubrication of bearings are included. In this section is also a discussion of lubricating closures and seals. Section V contains prices and weights.

The manual is available to executives and engineers responsible for bearing selection or maintenance and can be obtained by request on company letterhead.

Determination of Semi-Elliptic Laminated Springs

SAL COLACUORI
Fort Wayne, Ind.

The chart on the following page will facilitate the selection of a semi-elliptic chassis spring having graduated leaves. The chart is a graphic interpretation of the following formulas:

$$R = \frac{\Sigma I \times 32E}{K \times L^3} \qquad (1)$$

$$I = \frac{WT^3}{12} \qquad (2)$$

When:

R = Rate of deflection in lb. per in.
E = Modulus of elasticity = 28,000,000 lb. per sq. in.
K = Constant for semi-elliptic springs = 0.9
L = Full length of spring in in.
W = Width of leaves in in.
T = Thickness of leaves in in.
I = Moment of inertia

To construct the chart, Equation (1) is solved for the sum of the moments of inertia (ΣI) required for a spring of a given capacity. In Equation (1) E and K are constant, leaving L, R and (ΣI) to contend with. Plotting the rate of deflection (R) in lb. per in. as the abscissa and (ΣI) as the ordinates, L^3 is found for any set of values of R and I. From any point on the lines representing the length of the spring the sum of the moments of inertia (ΣI) required for a spring could be found if the values were shown. However, knowledge of these values is not necessary for the use of the spring chart and would only tend to complicate the chart if such values were shown.

The sum of the moments of inertia, (ΣI), now being known, the next step is to find the value of I for one leaf, which will be the value of I to use in Equation (2). Therefore, lines for the number of leaves are drawn, these lines dividing (ΣI) by the number of leaves and thus giving I for one leaf. With the value of I solved, the width of the leaves is now plotted. Dividing graphically by W gives I for a leaf one inch wide. The equation $I = WT^3/12$ can now be written $12(I) = T^3$, where I is the known moment of inertia for a leaf one inch wide. This value, then, equals T^3. The parabolic curve at the upper right-hand part of the chart is plotted to solve T^3 for T graphically.

The accompanying example shows how to use the chart. Starting with the rate of deflection (R = 1200) in lb. per in. read straight up to the length of spring (L = 55 in.). Cross horizontally to the line representing the number of leaves (6 leaves) then vertically to the line in the upper section of the chart corresponding to the width of the spring (W = 5 in.). From this point follow the horizontal line to the parabolic curve. The figure, 0.4375 in., directly above this last intersection, designates the thickness of each leaf in the spring. The spring has 1200 lb. per in. rate of deflection, is 55 in. from eye to eye, has 6 leaves, 5 in. wide and each leaf is 0.4375 in. thick.

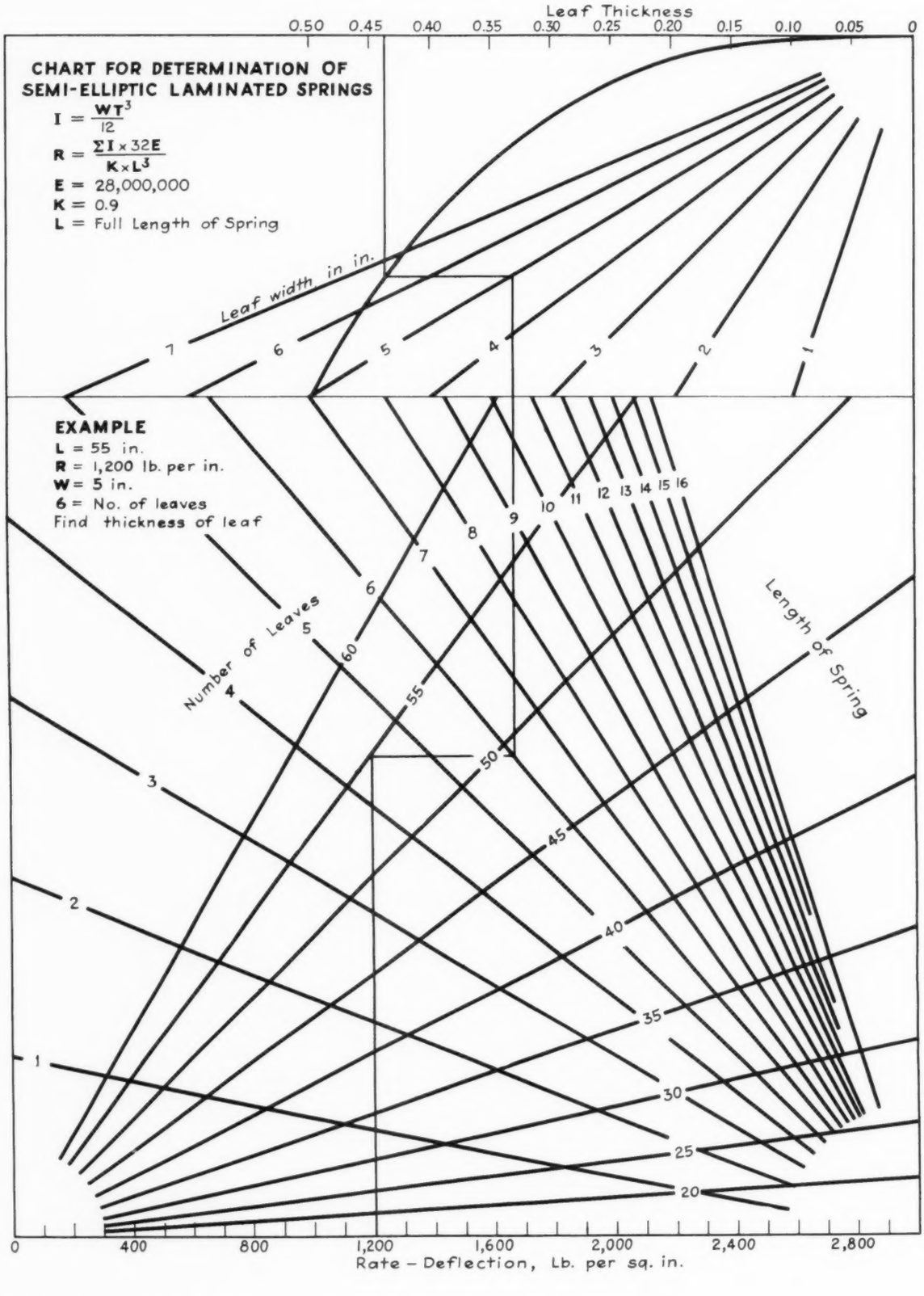
To find the safe load on the spring after the other values have been established from the chart:

$$S = \frac{4DET}{L^2} \text{ or } D = \frac{SL^2}{4ET}$$

Where:

S = Unit fiber stress in lb. per sq. in.
D = Amount of deflection in in.
E = Modulus of elasticity
T = Thickness of leaves in in.
L = Full length of spring in in.

The allowable working fiber stress will vary with the material used. Usually one-third of the elastic limit may be considered a safe working stress. For example, if the elastic limit is 180,000 lb. per sq. in., the safe unit stress will be 60,000 lb. per sq. in. Substituting this latter value for S in the formula, the amount of deflection D can then be solved. D multiplied by R (rate of deflection in lb. per in.) will give the full load capacity of the spring. In practice the spring may be stressed to two-thirds of the elastic limit, but only as an emergency load on the spring and not as the working stress.


Leaf Thickness
0.50 0.45 0.40 0.35 0.30 0.25 0.20 0.15 0.10 0.05 0
CHART FOR DETERMINATION OF SEMI-ELLIPTIC LAMINATED SPRINGS
I = WT³/12
R = ΣI × 32E / K × L³
E = 28,000,000
K = 0.9
L = Full Length of Spring
Leaf width in in.
7 6 5 4 3 2 1
EXAMPLE
L = 55 in.
R = 1,200 lb. per in.
W = 5 in.
6 = No. of leaves
Find thickness of leaf
Number of Leaves
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16
Length of Spring
60 55 50 45 40 35 30 25 20
0 400 800 1,200 1,600 2,000 2,400 2,800
Rate – Deflection, Lb. per sq. in.

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